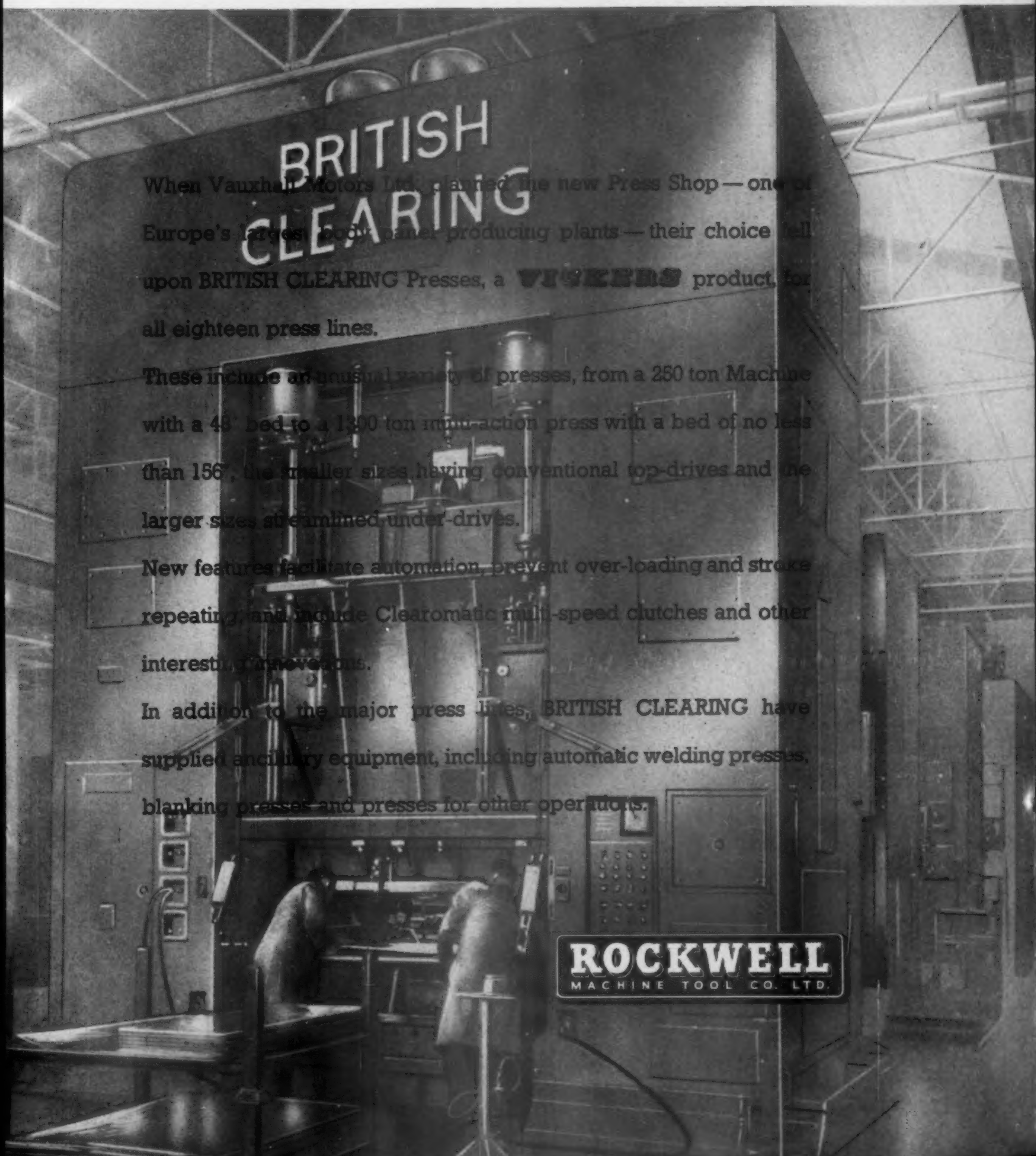
168

AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

Vol. 48 No. 7 JULY 1958 PRICE: 3s. 6d

One Chapter of the VAUXHALL Story



When Vauxhall Motors Ltd. planned the new Press Shop — one of Europe's largest body panel producing plants — their choice fell upon BRITISH CLEARING Presses, a VICKERS product, for all eighteen press lines.

These include an unusual variety of presses, from a 250 ton Machine with a 48" bed to a 1300 ton multi-action press with a bed of no less than 156", the smaller sizes having conventional top-drives and the larger sizes streamlined under-drives.

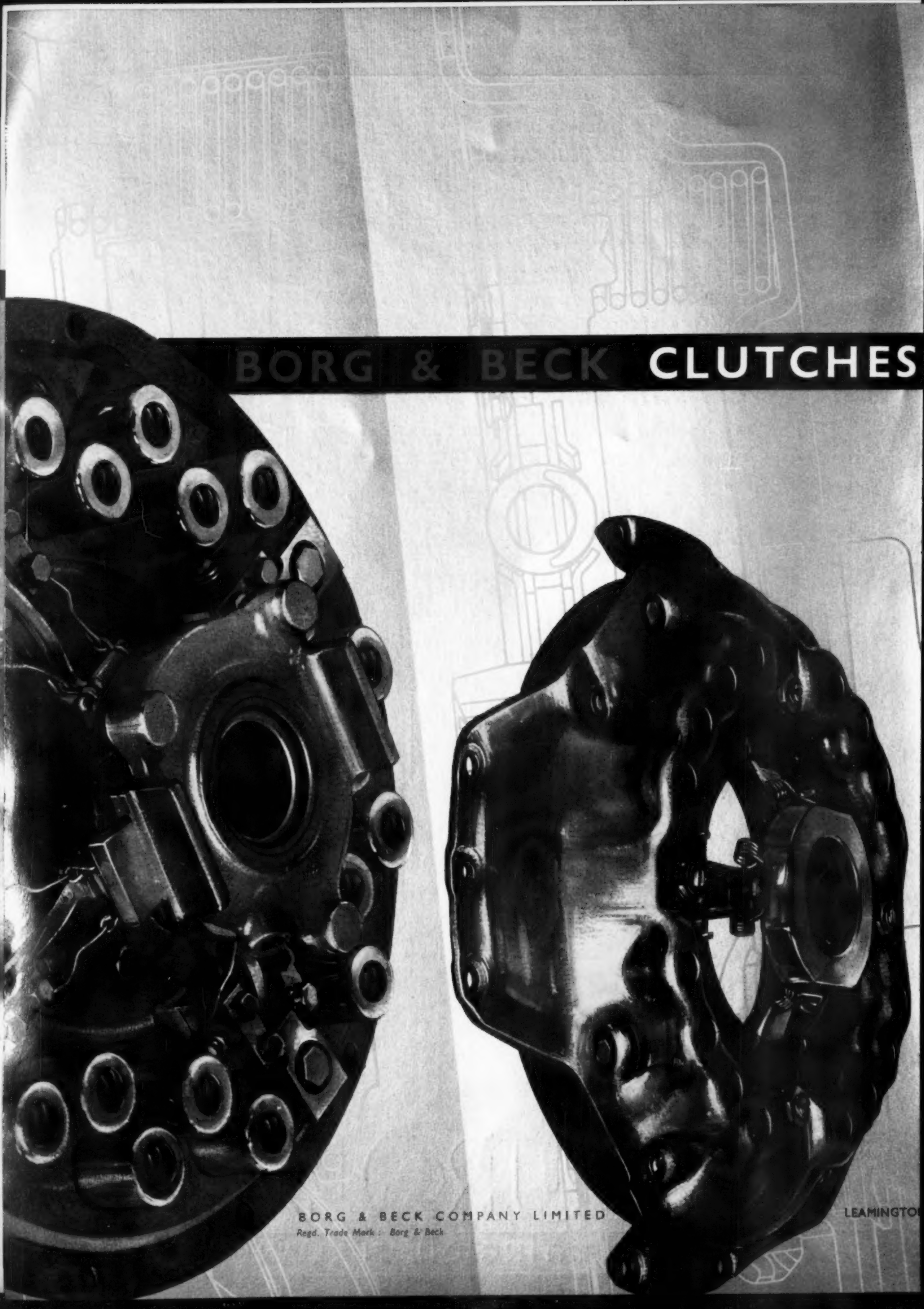
New features facilitate automation, prevent over-loading and stroke repeating, and include Clearomatic multi-speed clutches and other interesting innovations.

In addition to the major press lines, BRITISH CLEARING have supplied ancillary equipment, including automatic welding presses, blanking presses and presses for other operations.

ROCKWELL
MACHINE TOOL CO. LTD.

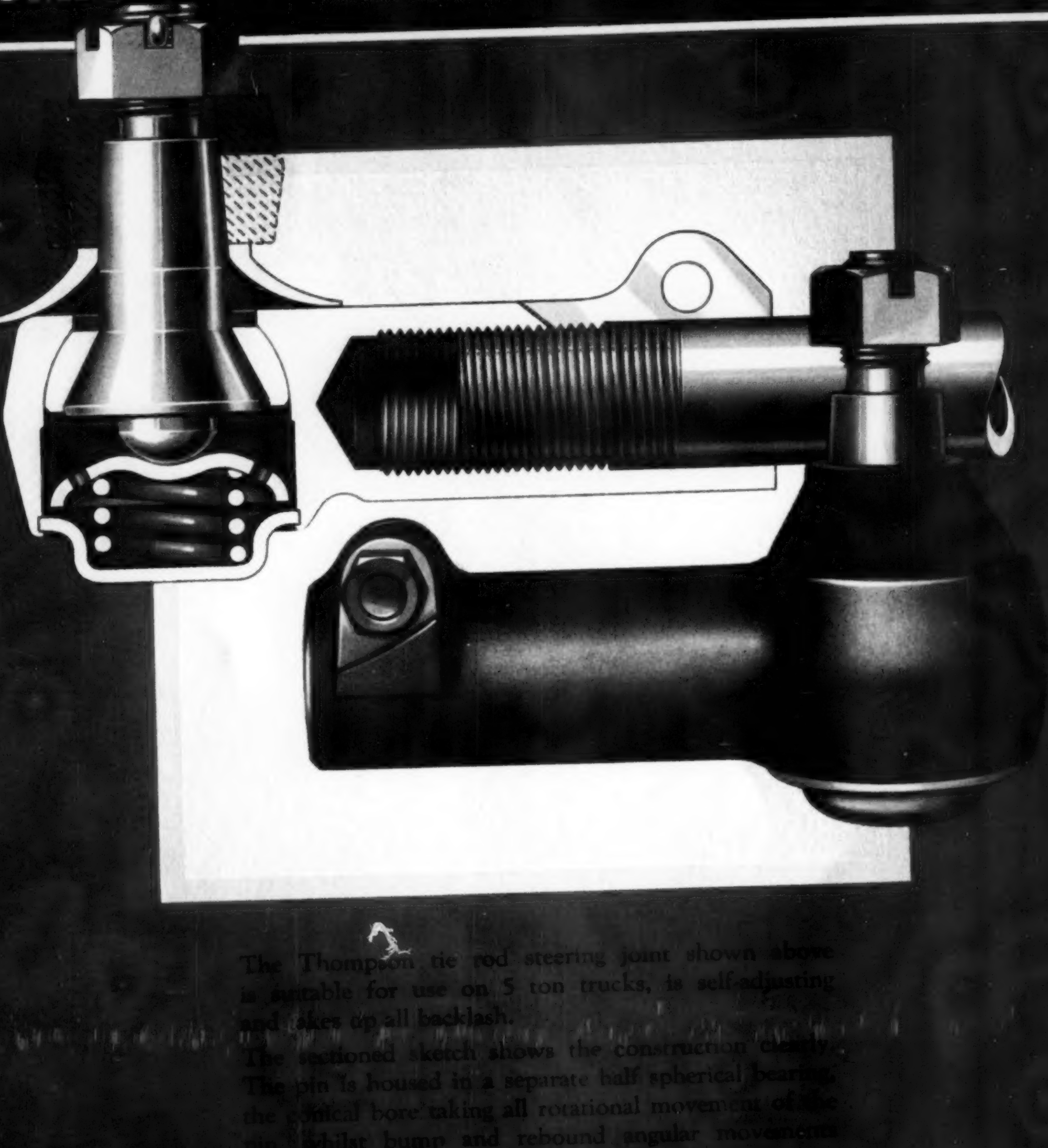
Part of one of Europe's Largest

Body Panel Producing Plants


BORG & BECK CLUTCHES
BORG & BECK COMPANY LIMITED
LEAMINGTO
Regd. Trade Mark : Borg & Beck


FOR ALL REQUIREMENTS
WARWICKSHIRE
ENGLAND

THOMPSON TIE-RODS: *DUAL TRUCK TYPE*

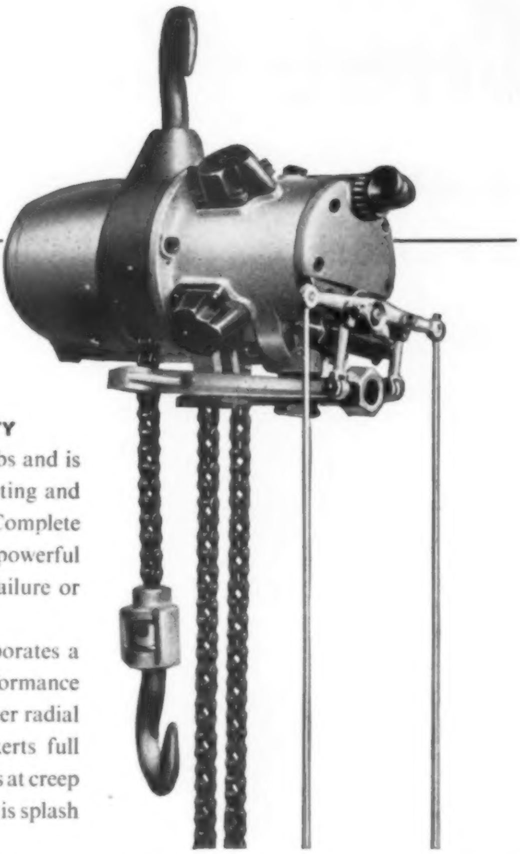


The Thompson tie rod steering joint shown above is suitable for use on 5 ton trucks, is self-adjusting and takes up all backlash.

The sectioned sketch shows the construction clearly. The pin is housed in a separate half spherical bearing, the conical bore taking all rotational movement of the pin, whilst bump and rebound angular movements are taken by the socket body and spherical bearing.

The range of Thompson dual truck type steering joints are made to suit vehicles up to 7 tons.

NEW LIGHTWEIGHT AIR MOTOR HOIST FROM ATLAS COPCO



The well-known Atlas Copco MLT range includes air motor hoists ranging up to 10 tons capacity. The newest addition to the range is the smallest and lightest of all—the MLT 11 KR, with a lifting capacity of a ¼ ton.

PERFECTLY CONTROLLED SPEED AND COMPLETE SAFETY

Despite its high capacity the MLT 11 KR weighs only 40 lbs and is easily portable. The 4-cylinder air motor gives smooth starting and perfectly controlled speed from creep to 35 feet per minute. Complete safety is assured by the upper and lower limit stops and a powerful automatic brake which prevents racing in case of supply failure or when throttle is closed.

The MLT 11 KR is equipped with roller chain and incorporates a number of design features which contribute to its high performance and the safety and economy of its operation. The four cylinder radial type piston air motor has precision made pistons, and exerts full torque regardless of throttle opening, permitting smooth starts at creep speed. All shafts are provided with roller bearings. The motor is splash lubricated.

PRINCIPAL DATA	Capacity at 85 psi	Operating speed at full load: lifting	lowering	Standard height of lift	Air consumption at full load	Weight	Hose
	¼ ton	35 ft/min	100 ft/min	10 ft	0.8 cu ft/ft	40 lb	⅜″

WIDE FIELD OF APPLICATION

The fine throttle control permits effortless and safe lifting, lowering and positioning of loads. For this reason the new hoist is already in demand for servicing machine tools and assembly lines, in foundries, press rooms and shipping rooms, etc. It lends itself particularly well for work in pickling plants and other acid or salt-laden atmospheres, or where spark hazards must be eliminated. On shipboard this light portable hoist is invaluable for speeding up all kinds of repair and maintenance work.

A COMPLETE RANGE OF COMPRESSED AIR EQUIPMENT

Atlas Copco manufactures portable and stationary compressors, rock-drilling equipment, loaders, pneumatic tools and paint-spraying equipment. Sold and serviced by companies or agents in ninety countries throughout the world.

Atlas Copco PUTS COMPRESSED AIR TO WORK FOR THE WORLD

Contact your local company or agent or write to Atlas Copco AB, Stockholm 1, Sweden, or Atlas Copco (Great Britain) Limited, Beresford Avenue, Wembley, Middlesex

T.11

your automatic choice

Battery of Herbert Auto-juniors at Moss Gear Co. Ltd.

Herbert No. 4 Auto-lathe at Moss Gear Co. Ltd., machining rear wheel hubs from medium carbon steel forgings.

For high-production repetition work with consistent accuracy.

Greatest output per foot of floor space at lowest labour cost per piece—one operator and one tool setter can keep from four to six machines in continual operation.

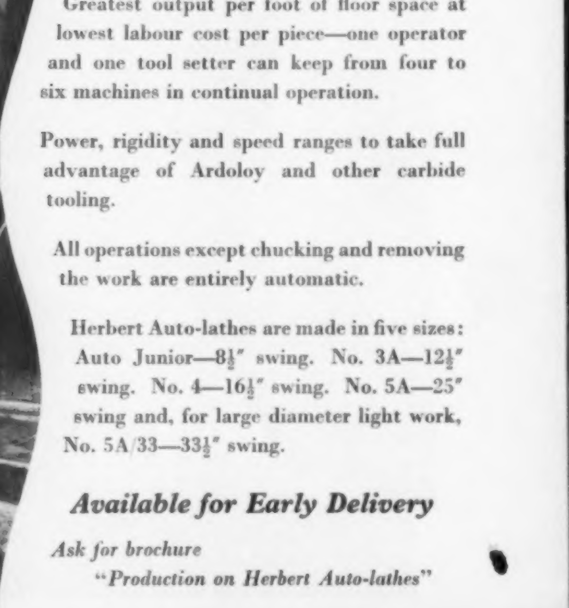
Power, rigidity and speed ranges to take full advantage of Ardoloy and other carbide tooling.

All operations except chucking and removing the work are entirely automatic.

Herbert Auto-lathes are made in five sizes: Auto Junior—8½" swing. No. 3A—12½" swing. No. 4—16½" swing. No. 5A—25" swing and, for large diameter light work, No. 5A/33—33½" swing.

Available for Early Delivery

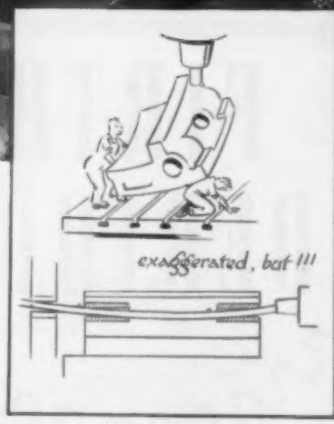
Ask for brochure
"Production on Herbert Auto-lathes"



ALFRED HERBERT LTD., COVENTRY

AD.417


DEVLIEG method ----
index table permits in-line
boring with short rigid
boring bars -----
outboard supports
eliminated.

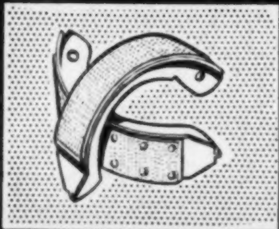

exaggerated, but !!!

6 GOOD REASONS

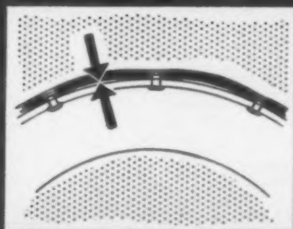
FOR FITTING GIRLING REPLACEMENT SHOES

1 Girling replacement shoes are inspected to the standard of original equipment

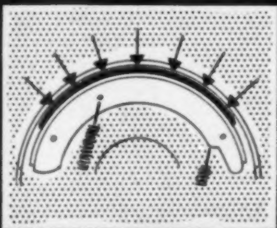
4 They save you garage time by being perfectly simple to fit



2 They are correctly riveted or bonded and precision ground to specified contours



5 They ensure that the lining fits tight to the shoe platform and so avoids this kind of gapping



3 They bed down quickly to full contact area and give highest efficiency braking in minimum time

6 They are your assurance that your vehicle braking is as good as new

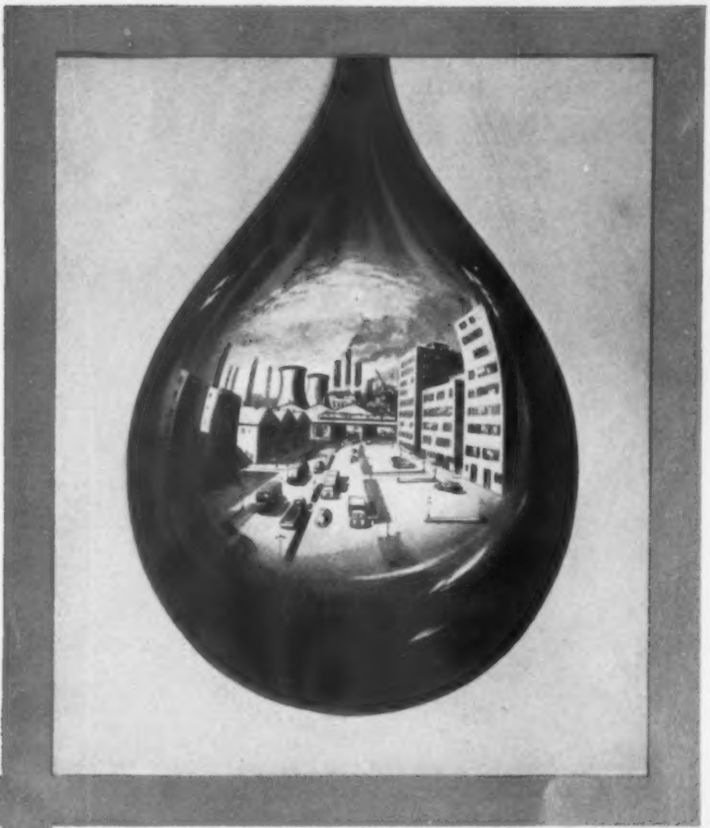
Don't reline—replace with
GIRLING FACTORY LINED
REPLACEMENT BRAKE SHOES

GIRLING

THE BEST BRAKES IN THE WORLD

GIRLING LIMITED · KINGS ROAD · TYSELEY · BIRMINGHAM 11

Lubrication tests at R&M smooth the flow of power



I understand Ransome & Marles make a special study of lubrication problems?

We do indeed. A bearing's performance is governed by correct lubrication as much as by correct design and materials. The cage location area is one of vital importance. The lubricant must also protect the highly finished surfaces from corrosion and help to dissipate any undue heat. To fulfil these requirements, we have a team of lubrication specialists at work.

What special lubrication problems do they have to deal with?

Our Technical Department study each case individually before making their recommendations. They may advise on grease for horizontal or vertical shafts, with the correct housing design and method of packing. If oil is used, the recommendation may involve lubrication by oil splash, a controlled oil level, drip feed or wick feed, and so on. All the time, of course, the various oil companies are developing new lubricants and these we subject to stringent tests before we recommend them for any application.

What about lubrication at ultra-high speeds?

Research into this is continuous. Operating speeds, loads and temperatures are steadily rising and every week brings fresh problems for solution. Aircraft and guided missiles are the obvious—but by no means the only—instances. Speaking generally we recommend lubrication by oil flood or vapour mist for the very highest speeds. We have built up extensive data on this and allied problems. If you get in touch with our Technical Department they'll be only too glad to talk lubrication with you.

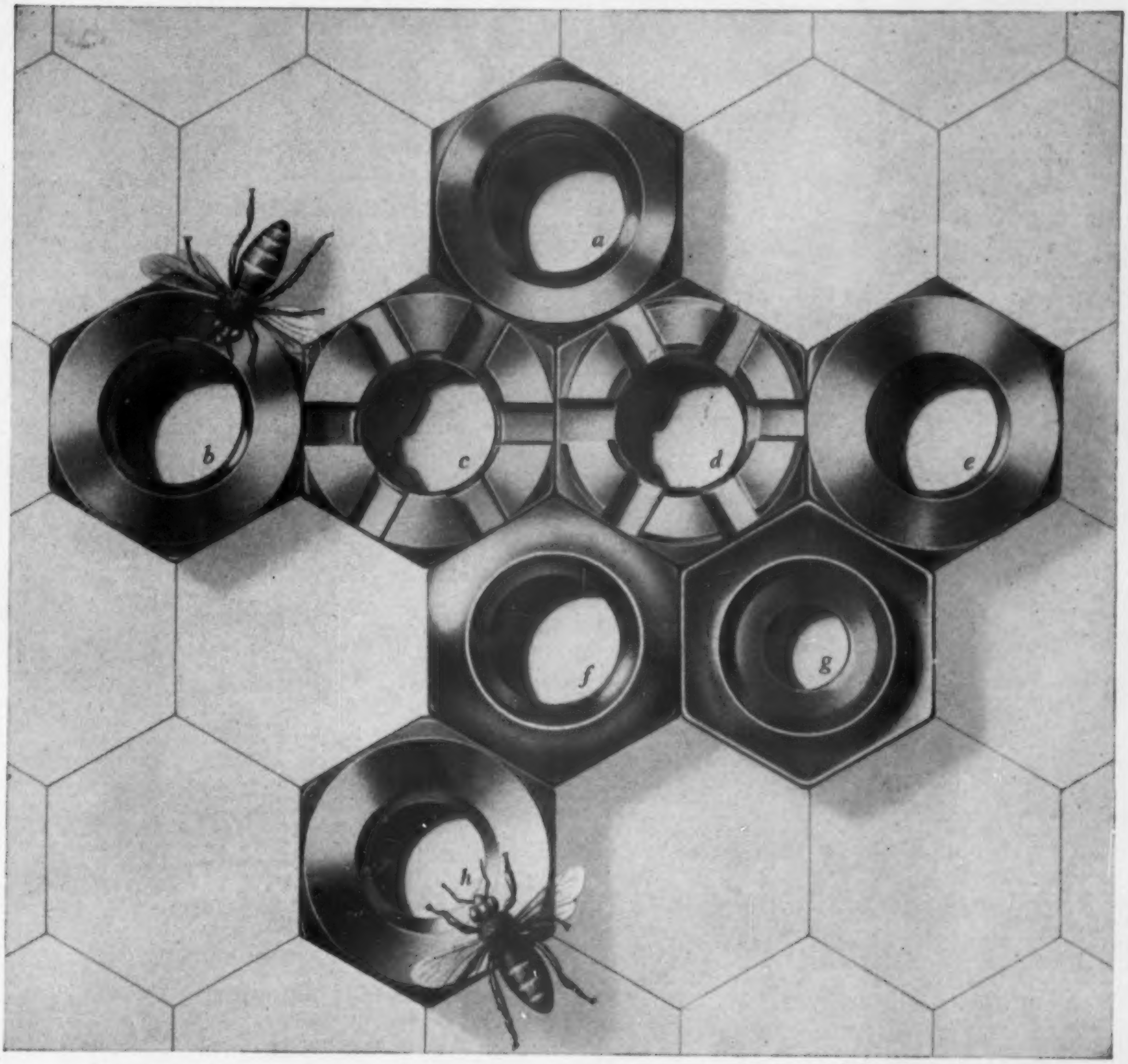
Ball & Roller R&M Bearings

RANSOME & MARLES BEARING CO. LTD.

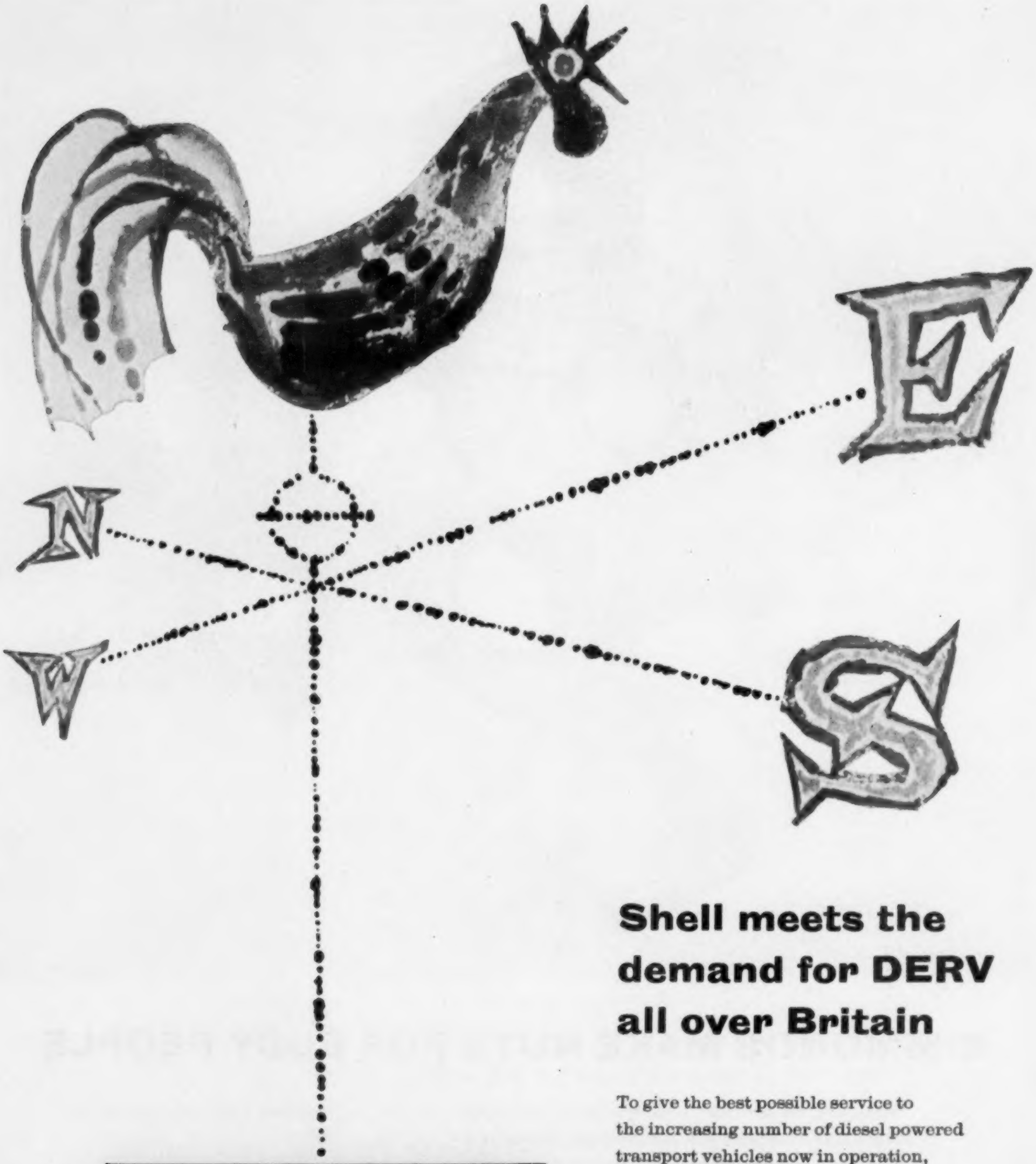
Newark-on-Trent England

Telephone Newark 456

Telex 37-306



a
b
c
d
e
f
g
h

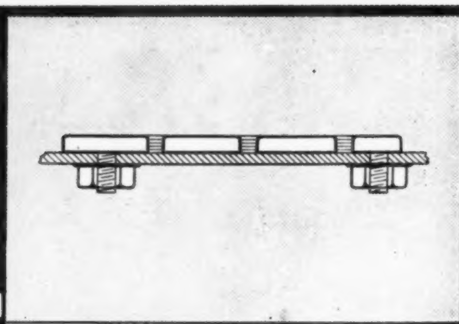


Shell meets the demand for DERV all over Britain

To give the best possible service to the increasing number of diesel powered transport vehicles now in operation, Shell have established a nation-wide network of agencies selling SHELL DERV. Wherever you send a diesel vehicle you can be sure that a SHELL DERV station will be close at hand. Your drivers can pay cash for SHELL DERV or by showing a Shell "Authority Card" they can refuel by a system of pre-arranged credit at any Shell agency.

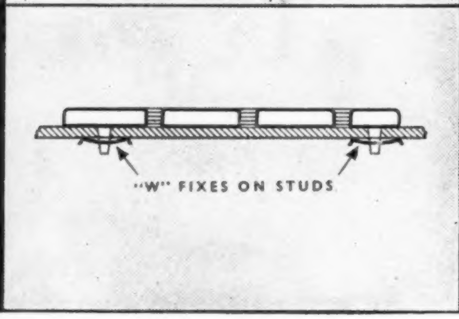
YOU CAN BE SURE OF SHELL DERV

The logical advance in Retaining



OLD WAY

Threaded studs were spot welded (or inserted if die-cast) to the nameplate. To hold this in position, washers and nuts were laboriously assembled with great difficulty especially on blind applications.



THE SALTER WAY

Studs are cast in with the nameplate, and the Fixes are simply pushed on, even on blind applications. The whole is firmly retained with an even spring tension.

save material–reduce assembly time–cut costs

When it's a question of assembling components in any engineering field, Salter Retainers are the answer. They replace nuts and bolts, screws, cotter pins, and eliminate expensive threading and machining operations. A large standard range is at your immediate disposal, and we should welcome the opportunity to assist in developing special retainers to solve your problems.

Send for the Salter Retainer catalogue — no designer is complete without it.

NEATER – MORE POSITIVE – PERMANENT RETAINING

SALTER

"TRUARC"

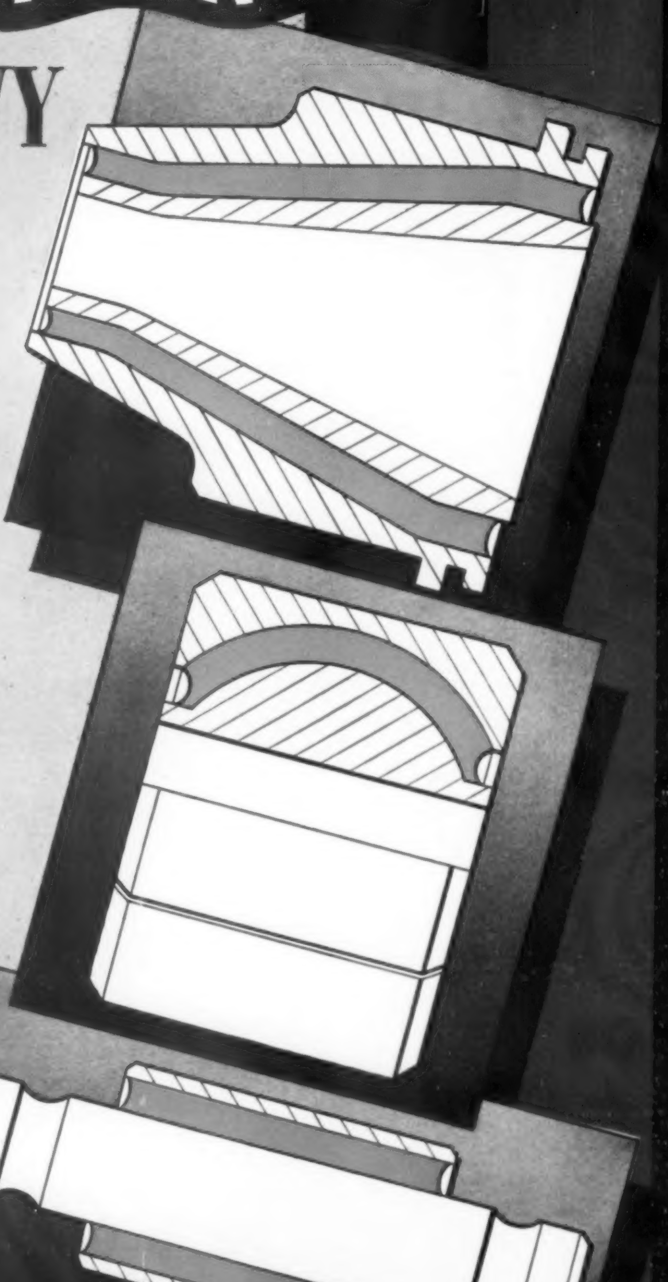
Circlips   Fasteners   Retainers   Fixes

Geo. Salter & Co. Ltd., West Bromwich · Spring Specialists since 1760

M-W.446

SUSPENSION UNITS

for the HEAVY VEHICLE



HEAVY-DUTY CONE BUSHES

Accommodating pivotal movements such as those in suspension linkage and spring pivots, these bushes are relatively free torsionally, adding but little to the spring stiffness. They are very stiff axially and radially and support heavy loads.

SPHERILASTIK BEARINGS

Permitting universal pivotal movements combined with large loads, Spherilastik bearings are being used with great success on a number of new suspension systems. Rubber is bonded to the spherical inner member and to the outer member which is in three segments. Force-fitting in the housing pre-loads the rubber to increase life and load capacity.

ULTRA-DUTY SHACKLE PINS

These are made in two types, one as illustrated and the other a still heavier duty type. They are giving amazingly long mileages in spring shackles on heavy vehicles.

Whatever your suspension, transmission or vibration problems may be, our engineers are always at your disposal.

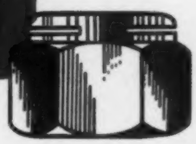
METALASTIK

METALASTIK LTD., LEICESTER

Photograph by courtesy of Ford Motor Co. Ltd.

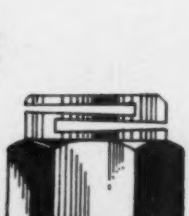
FORD consistently rely on

PHILIDAS Self-locking Nuts



Under conditions of extreme temperatures or vibration these locknuts

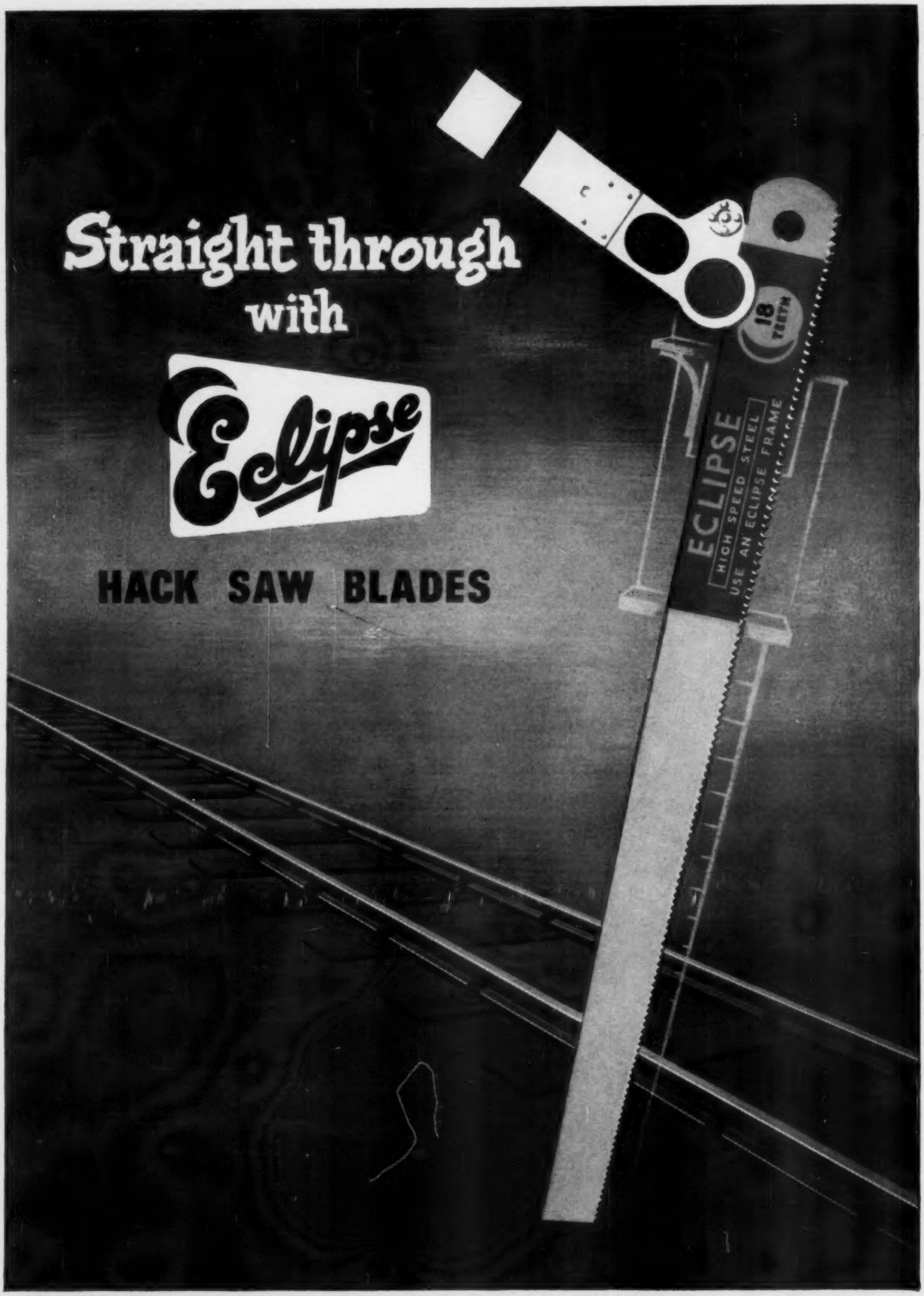
YIELD ONLY TO A SPANNER



Self-locking for the Automobile, Aircraft, Electrical and Engineering Industries. Wheel Nuts, Anchor Nuts, Strip Nuts, etc.

PHILIDAS DIVISION OF WHITEHOUSE INDUSTRIES LTD., FERRYBRIDGE, YORKSHIRE. Telephone: KNOTTINGLEY [illegible]
London Office: 44 Hertford Street, W.1. Telephone: Legation [illegible] Telex: [illegible] Associate Company: [illegible]


THE MOST COMPREHENSIVE RANGE IN THE COUNTRY!
Enots
LUBRICATING EQUIPMENT
The advantages of centralised systems for bearing lubrication are fully appreciated as an aid to production. Enots comprehensive range of lubrication equipment is adaptable to every industry and is indeed saving time, money and effort in the modern factory by reducing the number of breakdowns, eliminating costly repairs and adding years of service to the machine.
send NOW!
Enots
ONE-SHOT CENTRALISED LUBRICATING SYSTEM
Send now for a complete set of fully informative booklets which are free on request.
BENTON & STONE LTD., ASTON BROOK STREET, BIRMINGHAM 6. Telephone: ASTon Cross 1905



'Eclipse' hacksaw blades and other tools are made by James Neill & Co. (Sheffield) Ltd. and are obtainable from all tool distributors

UH25

tomorrow's motorist

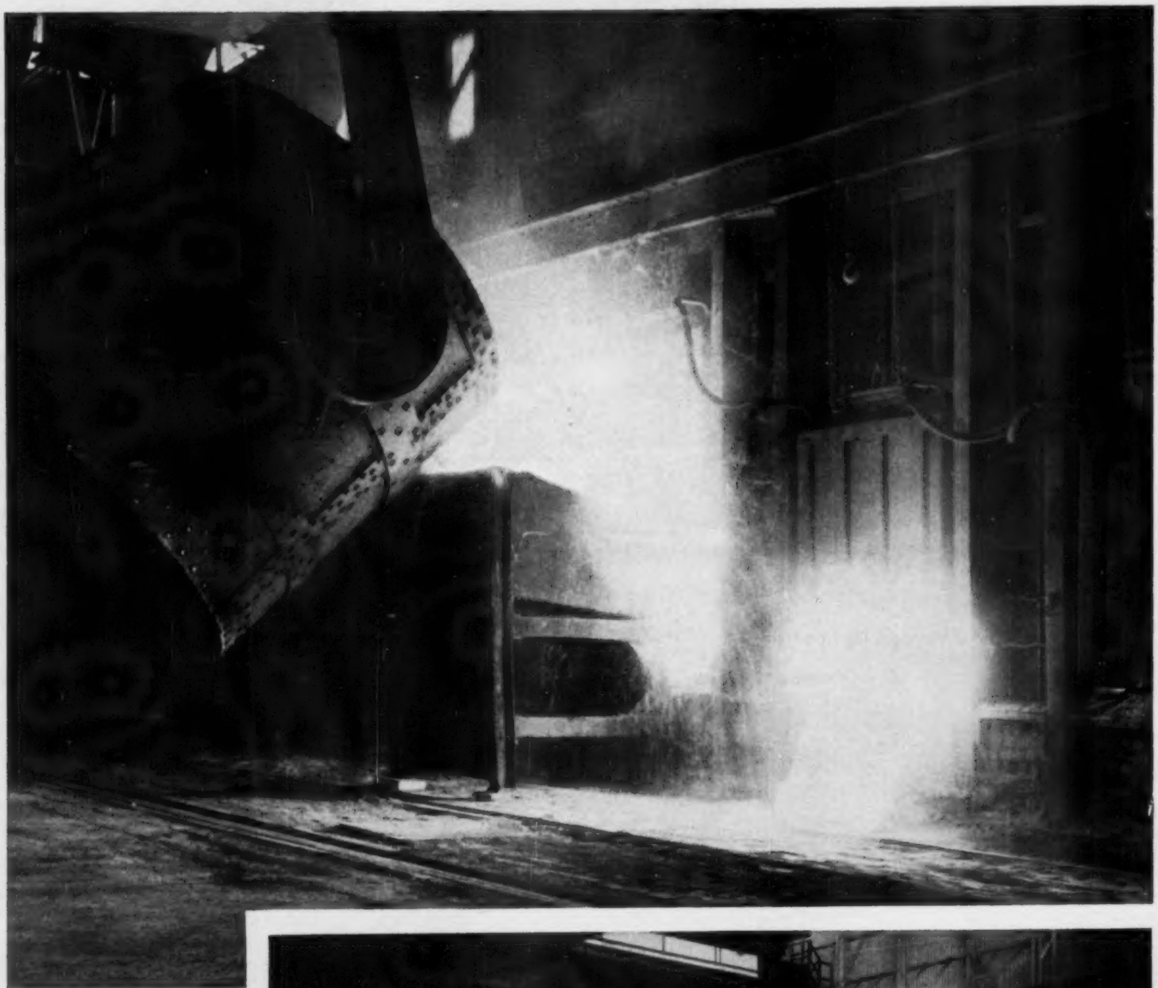
will insist on

Stainless Steel fittings

Issued by the makers of the famous 'Staybrite' *Stainless Steel*

FIRTH-VICKERS STAINLESS STEELS LTD. SHEFFIELD

One-third of Britain's SHEET STEEL



Molten iron being charged into an open hearth furnace at Abbey Works

The ingot stripper bay at Abbey Works

ENQUIRIES

to The Sales Manager:
Sheet and Plate—
Abbey Works,
Port Talbot,
Glamorgan.
Electrical Sheet—
Orb Works,
Newport,
Monmouthshire.

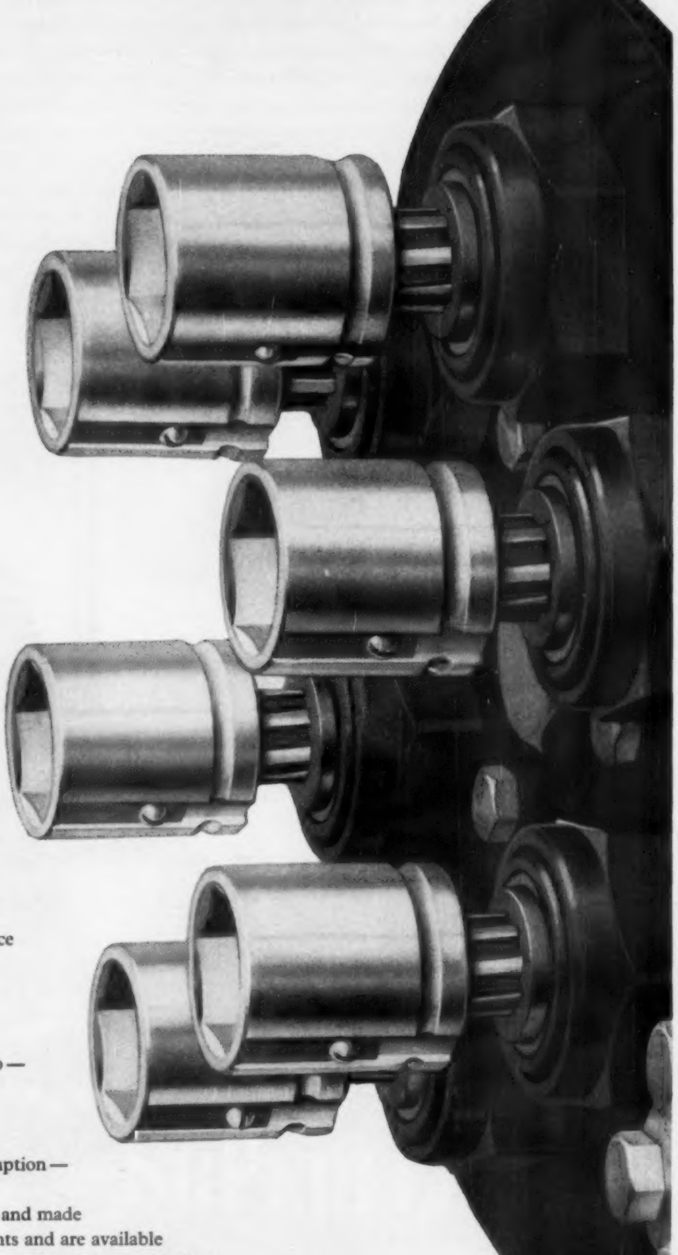
is made by

THE STEEL COMPANY OF WALES LIMITED

The slabbing mill motor room at Abbey Works

new *Silent* MULTIPLE nut runner units produce torques of

135 lb. ft.



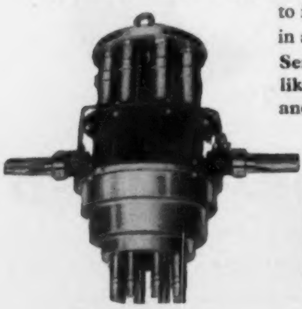
For years we've been making pneumatic multiple nut runner units with motors producing torques of up to 50 lb. ft. The *new* Desoutter motors produce torques of up to 135 lb. ft.—without consuming any extra air.

What's the big secret? A motor with a 2-speed gearbox. This motor works at fast speeds for running down nuts, at slow speeds for tightening up—and the change of speed is automatic.

Compared with any other multiple units these new Desoutter tools give *three* times the performance—or the same performance at one third the air consumption—with even greater uniformity of torque.

Desoutter Multiple Nut Runner Units are designed and made to individual requirements and are available in a wide range, for torques from 8 to 135 lb. ft.

Send for descriptive leaflet or, if you would like our representative to call, drop us a line and we'll arrange it.



8 spindle unit for differential housing assembly (Vauxhall Motors Ltd.)

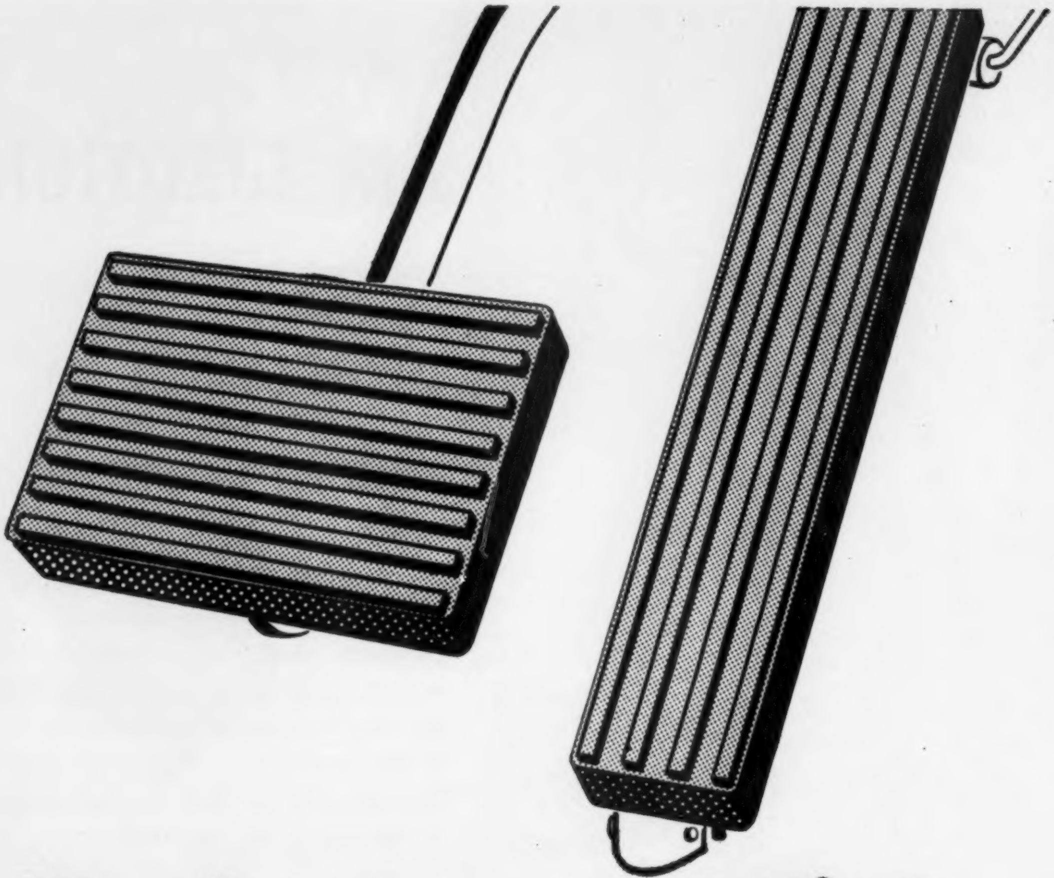
Desoutter 2-speed

multiple nut runner units

DESOUTTER BROS LIMITED, THE HYDE, LONDON, N.W.9. TEL: COLINDALE 6346 (5 LINES). GRAMS: DESPNUCO, HYDE, LONDON

CRC39M


CLAYFLEX
FLEXIBLE BEARINGS
The continuing improvement in the standard of performance of the modern road vehicle results in ever-increasing demands being made upon the components. The Clayflex Principle is a recognition of the need to provide a flexible bearing which, by design improvement, has the capacity to meet the more arduous conditions imposed upon it by everyday modern motoring. The success of the Clayflex Principle lies in the combination of two familiar techniques, chemical bonding and pressure bonding. The well-known advantages of the pressure bond obtained by controlled rubber deformation upon assembly are reinforced by the addition of chemical bonding of rubber to inner member. It goes without saying that the claim to success could not be made unless the technical improvement involved no additional cost. Clayflex Bearings are competitive.
A fully illustrated brochure is available, and requests for copies are welcomed from Chief Technicians and Buyers.
HOWARD CLAYTON-WRIGHT LTD
WELLESBOURNE · WARWICKSHIRE · ENGLAND
Phone: Wellesbourne 316/7/8 | Grams: Clatonrite Wellesbourne



Ferodo - Automatically First

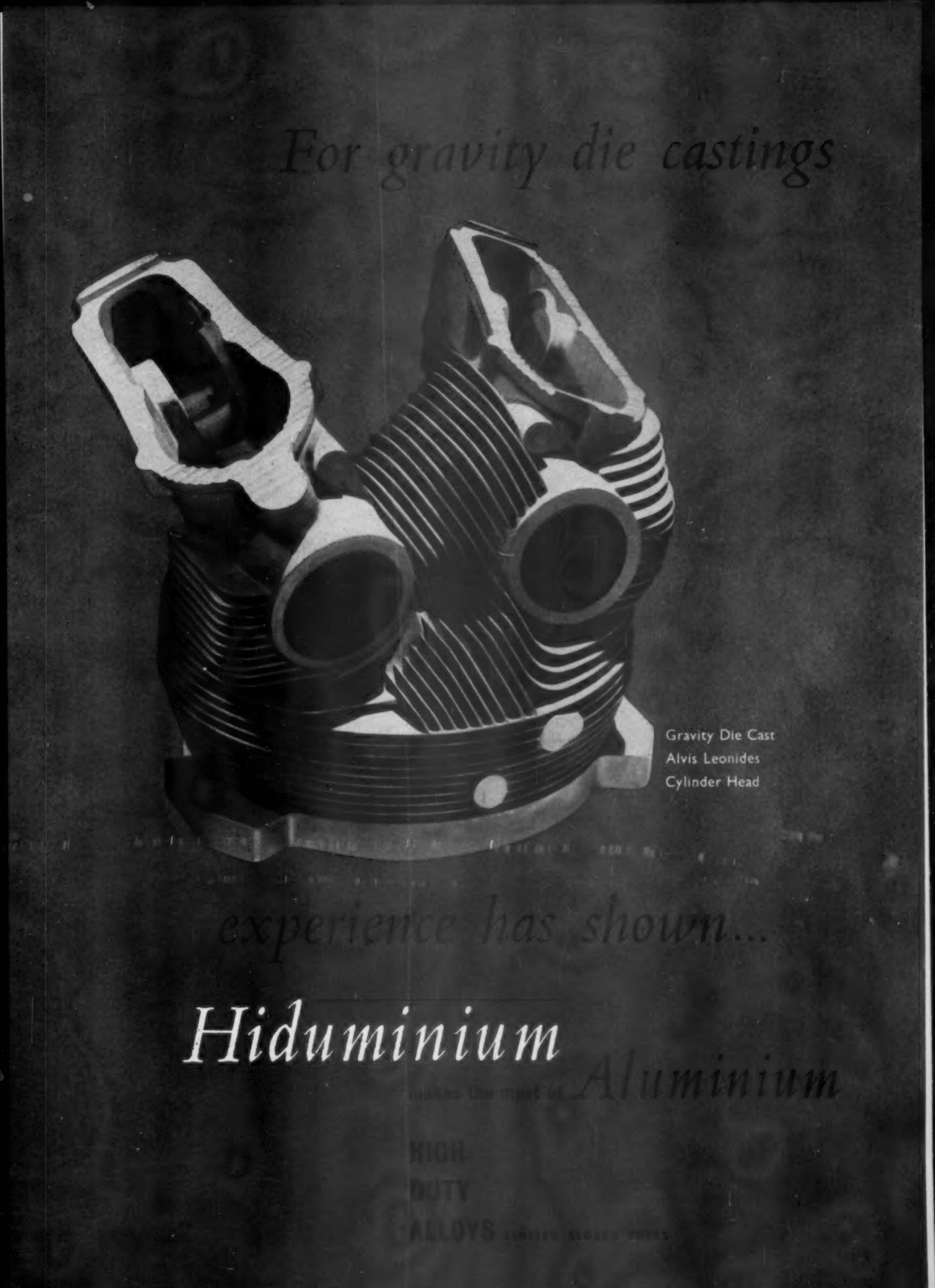
On ROLLS-ROYCE Automatic Transmission
BORG WARNER Automatic Transmission
FERLEC Clutch · NEWTON Drive
STANDRIVE

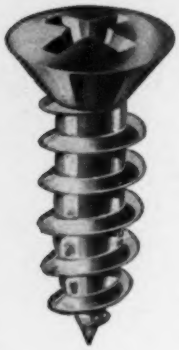
each system has its *own* particular quality
of Ferodo Friction Material

Ferodo technicians developed materials exclusively for each individual type of transmission whether oil immersed or dry. They are giving the smooth, long lasting trouble-free service that is associated with any component bearing the name . . . FERODO.

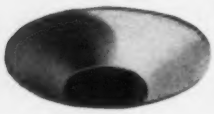
FERODO Friction Materials
for the automotive industry

FERODO LIMITED · CHAPEL-EN-LE-FRITH *A Member of the Turner & Newall Organisation*

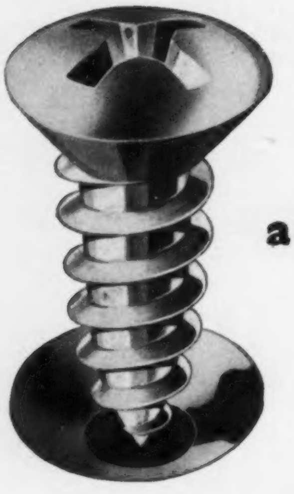

For gravity die castings
Gravity Die Cast
Alvis Leonides
Cylinder Head
experience has shown...
Hiduminium
makes the most of Aluminium
HIGH
DUTY
ALLOYS



there is



a plain



advantage

With steadily-rising labour costs, every non-essential operation along the production line assumes increasing significance. One of these recurring operations has been the precision thread-cutting of screw holes, which, in assembly, have (equally precisely) to be registered together. But the tapping, self-locating screw eliminates all this, and permits assembly by less-skilled labour. And the cross-recessed head version is designed for use with power-drivers—without a scratch of damage to worker or job. The tapping screw is demonstrably quicker—cheaper—stronger—safer than the conventional machine screw. Such plain advantages pay such handsome dividends!

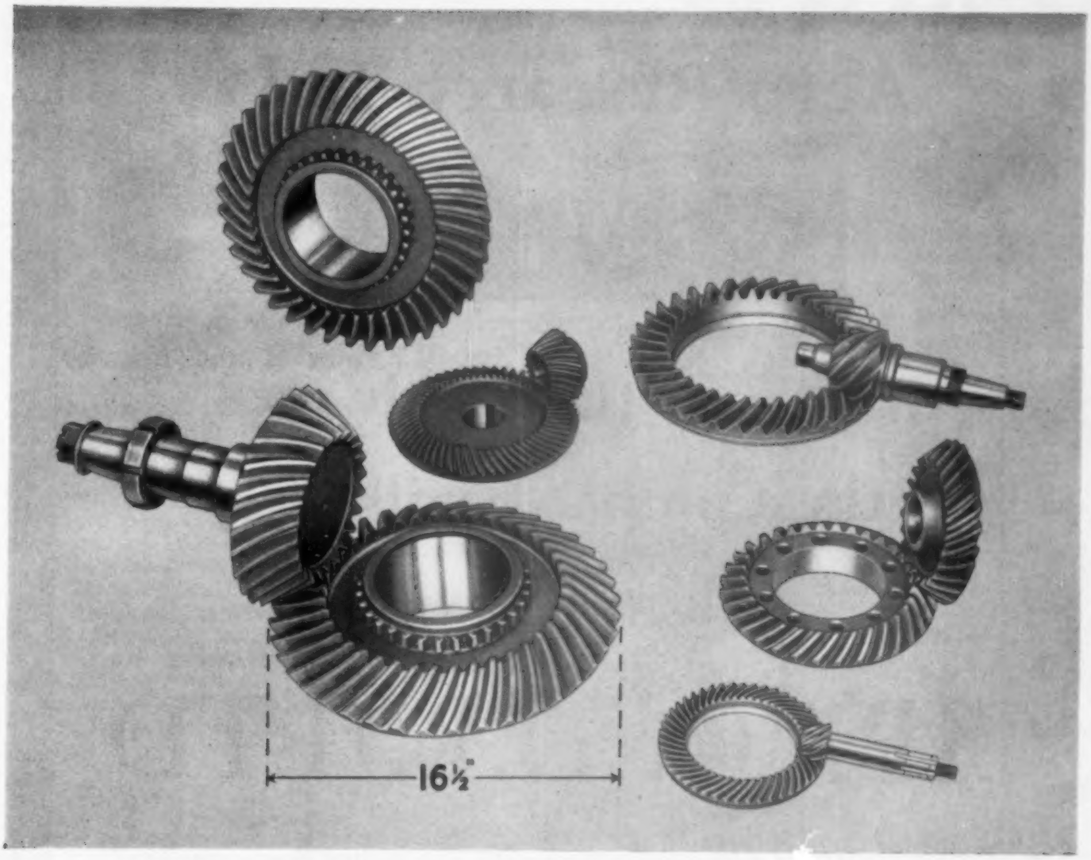
in tapping screws

Linread

LINREAD LTD · COX ST · BIRMINGHAM 3

Slotted and Phillips recessed head machine screws and tapping screws · High tensile hexagon headbolts and setscrews
Aircraft bolts, screws and rivets · Sems · Rivnuts · Custom designed cold forged fasteners · Alex Screwplugs

SPIRAL BEVEL GEARS . . .



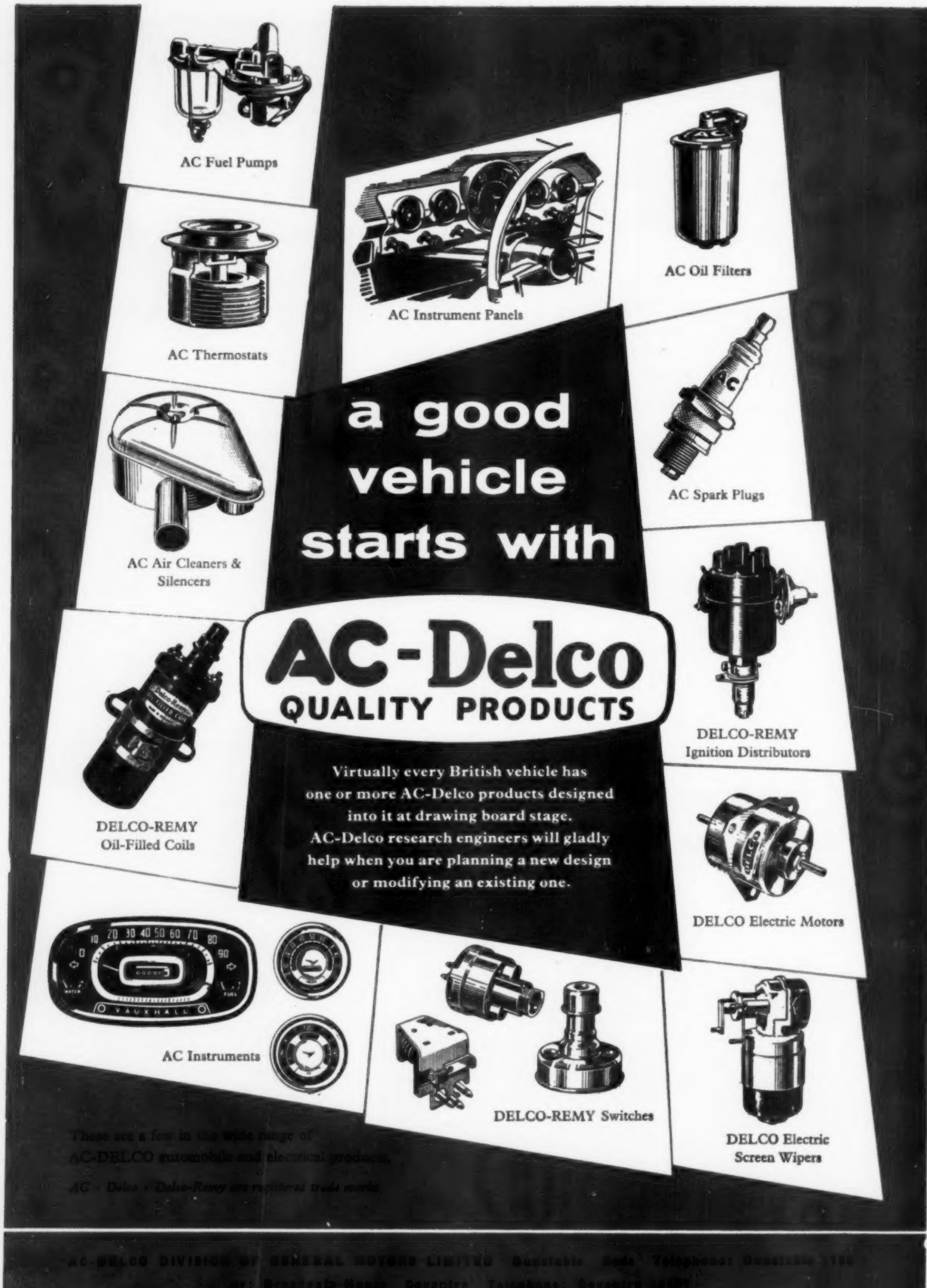
Amongst the largest producers of spiral bevel gears in Europe, we at The Moss Gear Co. have those many years of experience that do so much to smooth away design and production problems. You will find our technical staff happy to work with you—and their advice invaluable.

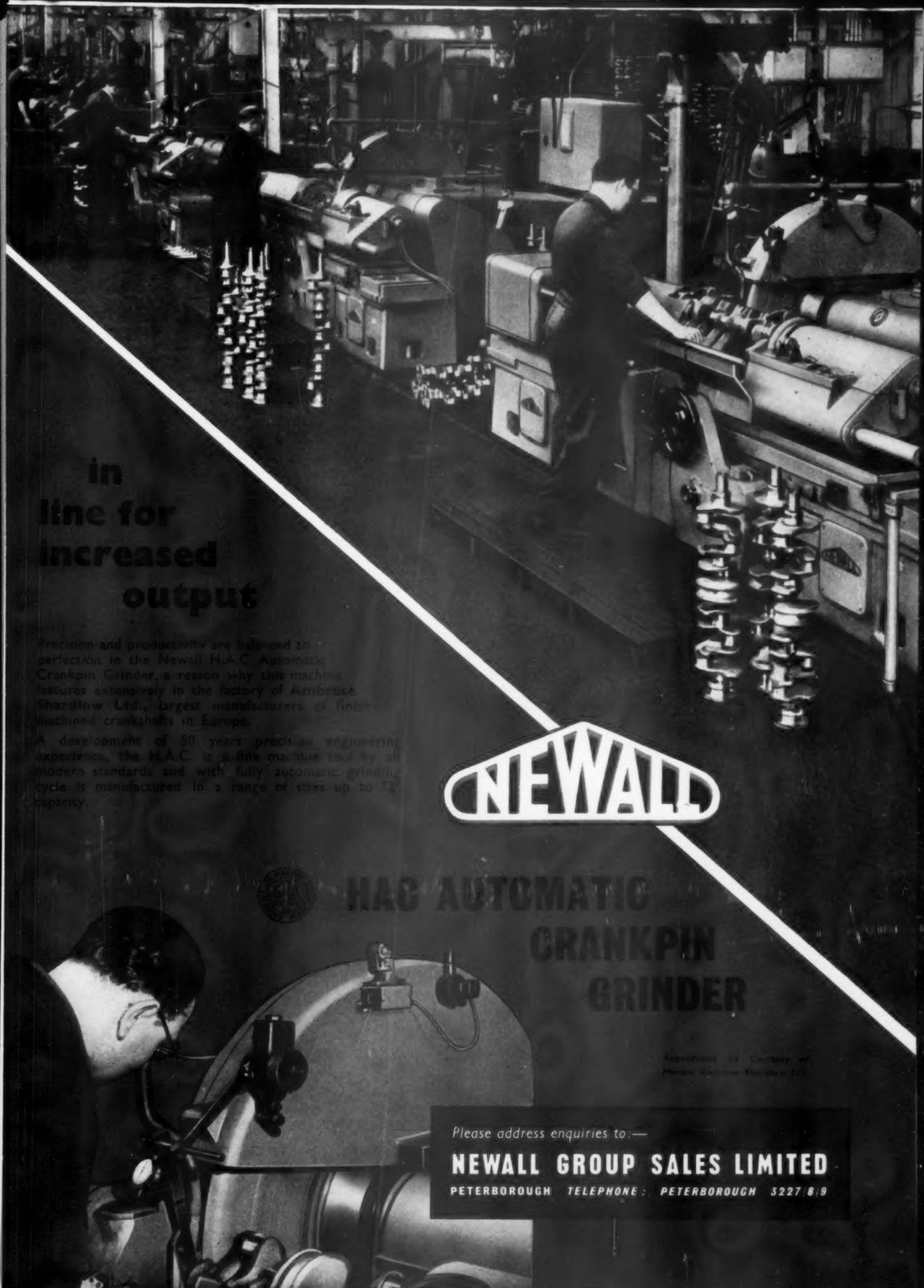
MOSS GEARS

measure up to all requirements

THE MOSS GEAR CO. LTD., CROWN WORKS, TYBURN, BIRMINGHAM, 24

Phone: ERDington 1661-6 Grams: 'Mosgear Birmingham'

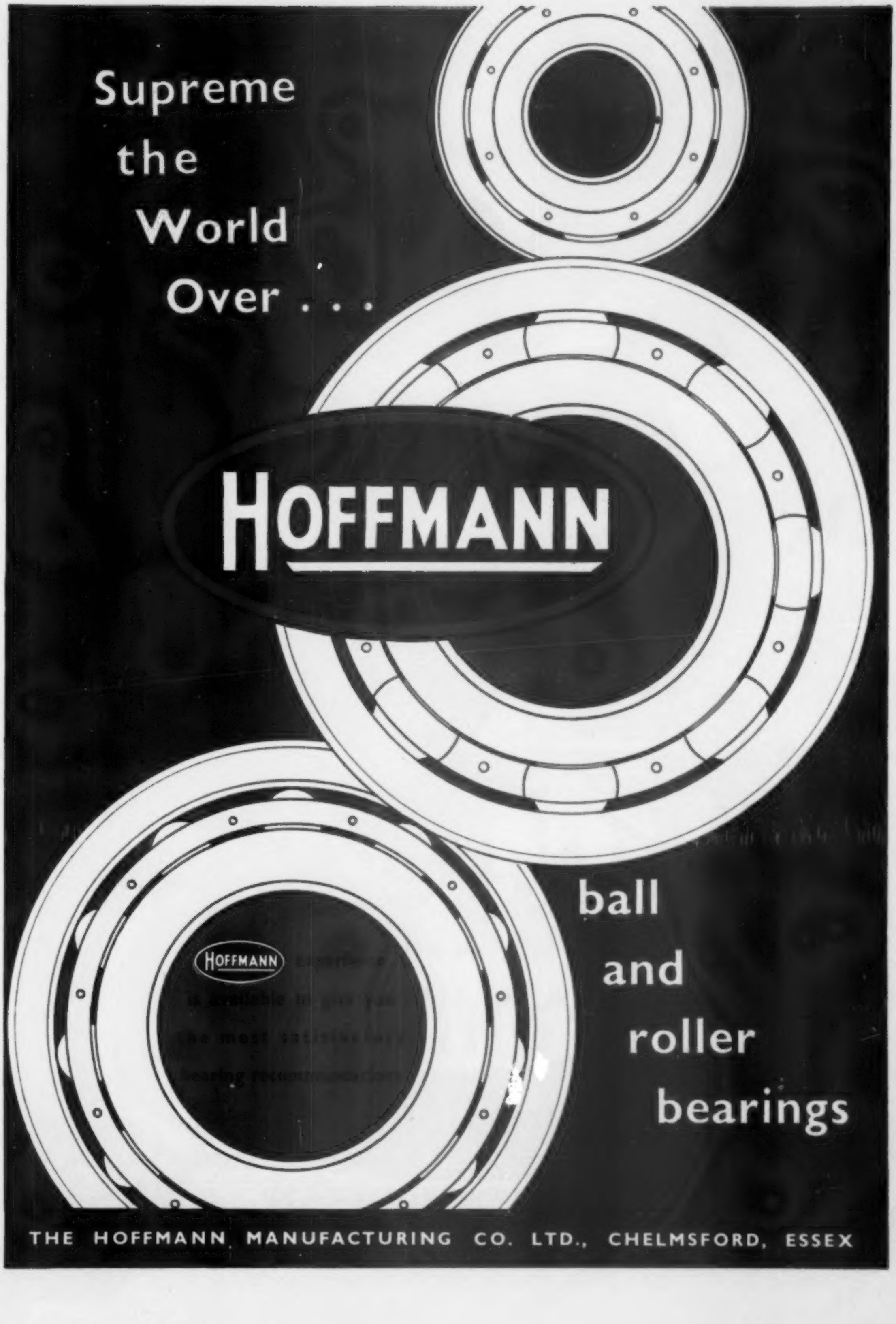

AC Fuel Pumps
AC Instrument Panels
AC Oil Filters
AC Thermostats
AC Spark Plugs
a good
vehicle
starts with
AC Air Cleaners & Silencers
AC-Delco
QUALITY PRODUCTS
DELCO-REMY Ignition Distributors
DELCO-REMY Oil-Filled Coils
Virtually every British vehicle has one or more AC-Delco products designed into it at drawing board stage. AC-Delco research engineers will gladly help when you are planning a new design or modifying an existing one.
DELCO Electric Motors
AC Instruments
DELCO-REMY Switches
DELCO Electric Screen Wipers
These are a few in the wide range of AC-DELCO automobile and electrical products.
AC · Delco · Delco-Remy are registered trade marks
AC-DELCO DIVISION OF GENERAL MOTORS LIMITED Dunstable Beds Telephone: Dunstable
Coventry Telephone: Coventry

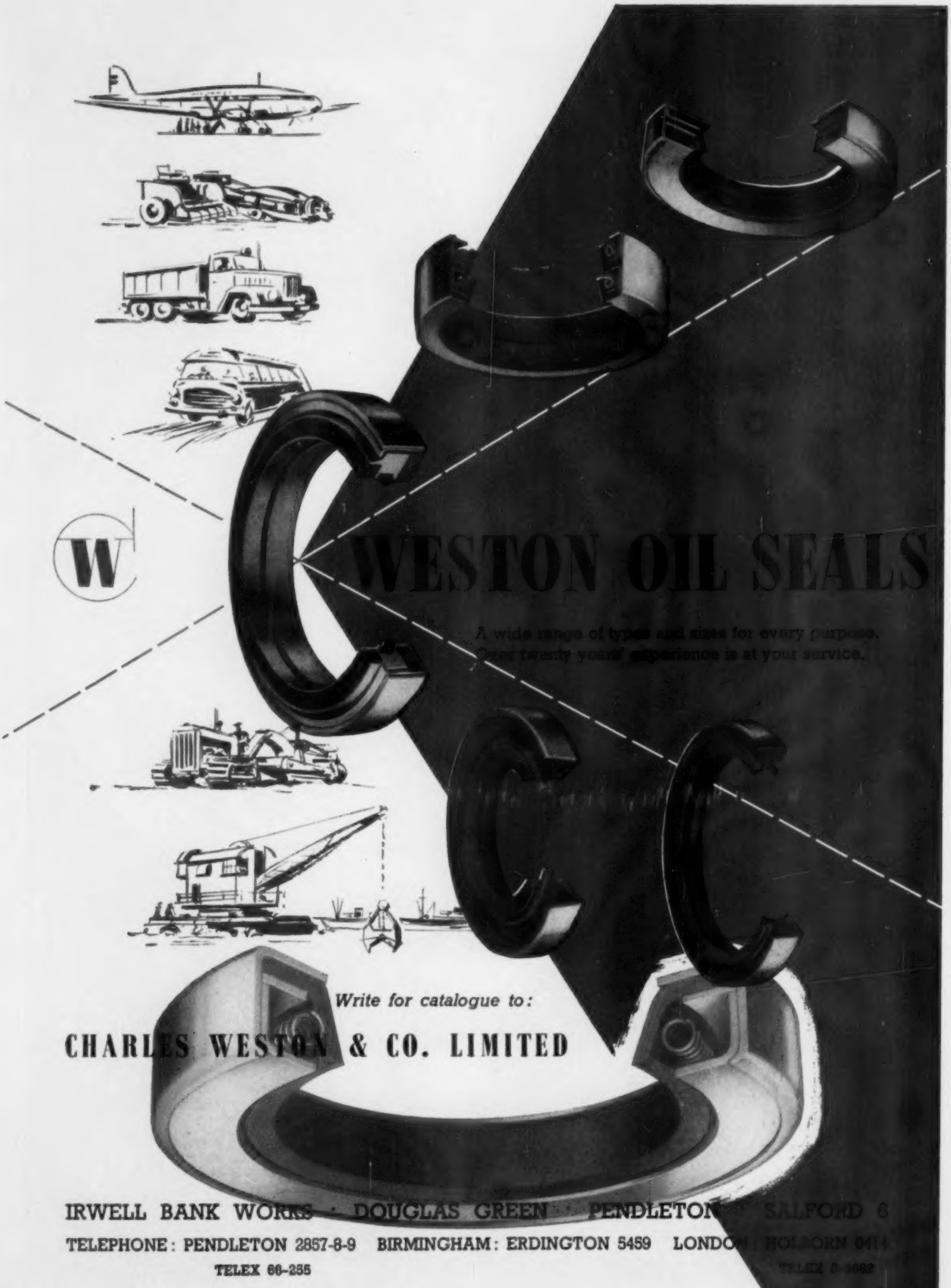

in line for increased output
Precision and productivity are balanced to perfection in the Newall H.A.C. Automatic Crankpin Grinder, a reason why this machine features extensively in the factory of Ambrose Shardlow Ltd., largest manufacturers of finished machined crankshafts in Europe.
A development of 50 years precision engineering experience, the H.A.C. is a fine machine tool by all modern standards and with fully automatic grinding cycle is manufactured in a range of sizes up to 72" capacity.
NEWALL
HAC AUTOMATIC CRANKPIN GRINDER
Please address enquiries to:—
NEWALL GROUP SALES LIMITED
PETERBOROUGH TELEPHONE: PETERBOROUGH 3227/8/9

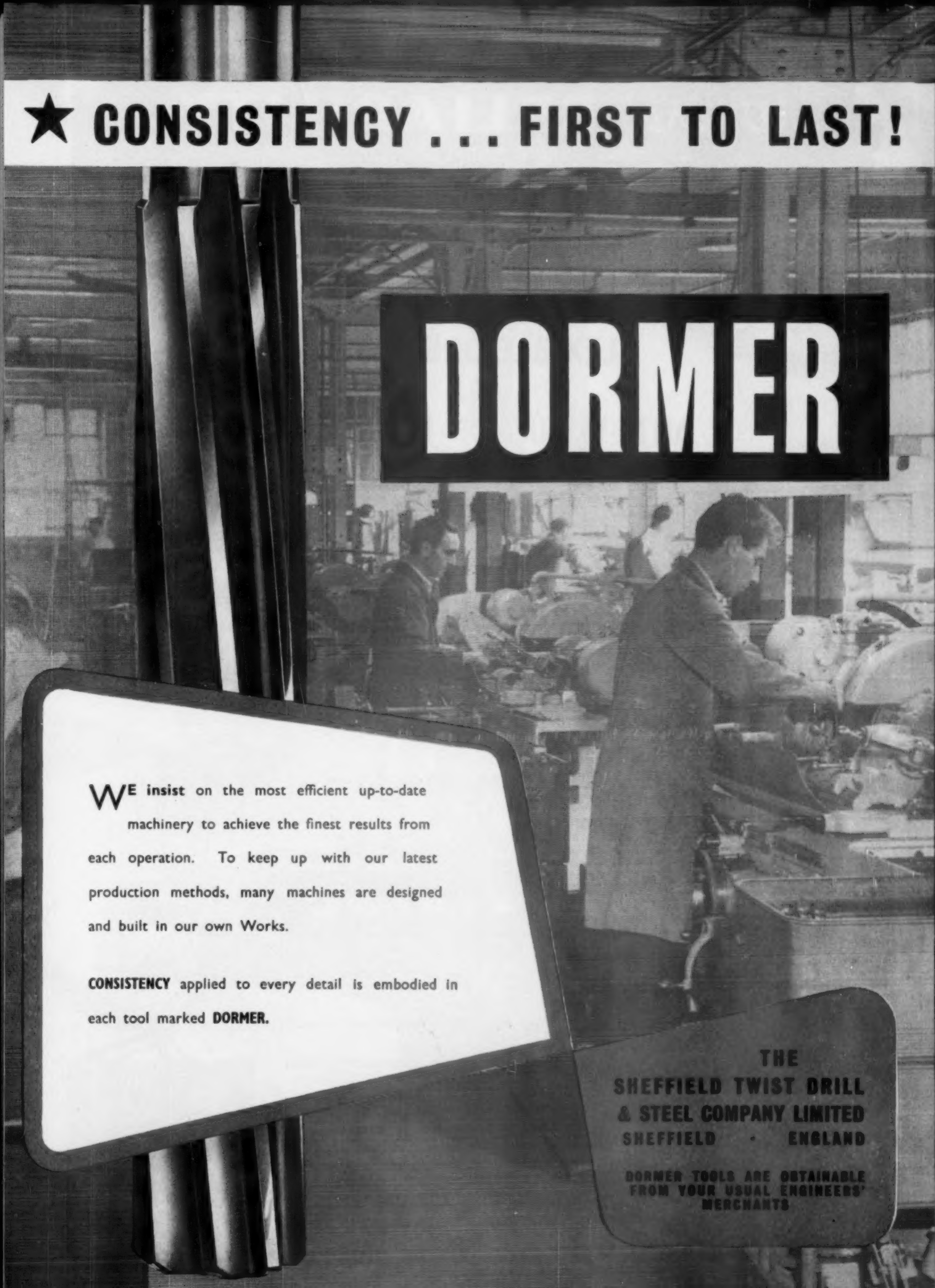

Morrisflex

vehicle springs
to
your
specification
KANTLINK
SPRING WASHERS
Will not link together into chains; in all sizes A.S.A. or S.A.E. medium type.
TOLEDO WOODHEAD SPRINGS LIMITED · SHEFFIELD 3

TW364


Supreme
the
World
Over . . .
HOFFMANN
HOFFMANN
ball
and
roller
bearings
THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX


WESTON OIL SEALS
A wide range of types and sizes for every purpose.
Over twenty years' experience is at your service.
Write for catalogue to:
CHARLES WESTON & CO. LIMITED
IRWELL BANK WORKS · DOUGLAS GREEN · PENDLETON · SALFORD 6
TELEPHONE: PENDLETON 2857-8-9 BIRMINGHAM: ERDINGTON 5459 LONDON: HOLBORN 0414
TELEX 66-255


★ CONSISTENCY . . . FIRST TO LAST!
DORMER
WE insist on the most efficient up-to-date machinery to achieve the finest results from each operation. To keep up with our latest production methods, many machines are designed and built in our own Works.
CONSISTENCY applied to every detail is embodied in each tool marked DORMER.
THE SHEFFIELD TWIST DRILL & STEEL COMPANY LIMITED
SHEFFIELD · ENGLAND
DORMER TOOLS ARE OBTAINABLE FROM YOUR USUAL ENGINEERS' MERCHANTS


Where RELIABILITY is paramount....
for
FLEXIBLE COUPLINGS
ANTI-VIBRATION MOUNTINGS
or
FLEXIBLE BEARINGS
the hallmark of course, is—
SILENTBLOC
REGISTERED TRADE MARK
At your disposal—collected over many years—is a wealth of technical information. Our technicians can be of most help to you when a project is on the drawing board; your problems are more than welcome.
Come to us at the drawing board stage —
SILENTBLOC LIMITED, MANOR ROYAL, CRAWLEY, SUSSEX

new insulation—light and strong

These simple-to-mix components give strong, lightweight rigid foams to provide an insulating material of particular value in the construction of refrigerated vehicles. Easily mixed on the site, the components produce the foam in the cavity to be filled. As this happens, a bond is formed with the wood or metal with which the foam is in contact adding strength and rigidity to the structure.

with Isocyanates

and POLYESTERS by ICI

Ask for details of **Daltolacs 21 & 22** and **Suprasec D**

Enquiries should be addressed to: I.C.I. Sales Development Department (Polyisocyanates), Ship Canal House, King Street, Manchester 2

D.966

IMPERIAL CHEMICAL INDUSTRIES LIMITED LONDON SW1 ENGLAND

Churchill

HBM

AUTOMATIC SIZING INTERNAL GRINDING MACHINE

CHURCHILL Automatic Sizing Internal Grinding Machines are renowned for their accuracy and speed of production. The extensive experience gained from supplying very many machines has been fully utilised in the Model 'HBM' to ensure highly efficient operation over a wide range of work.

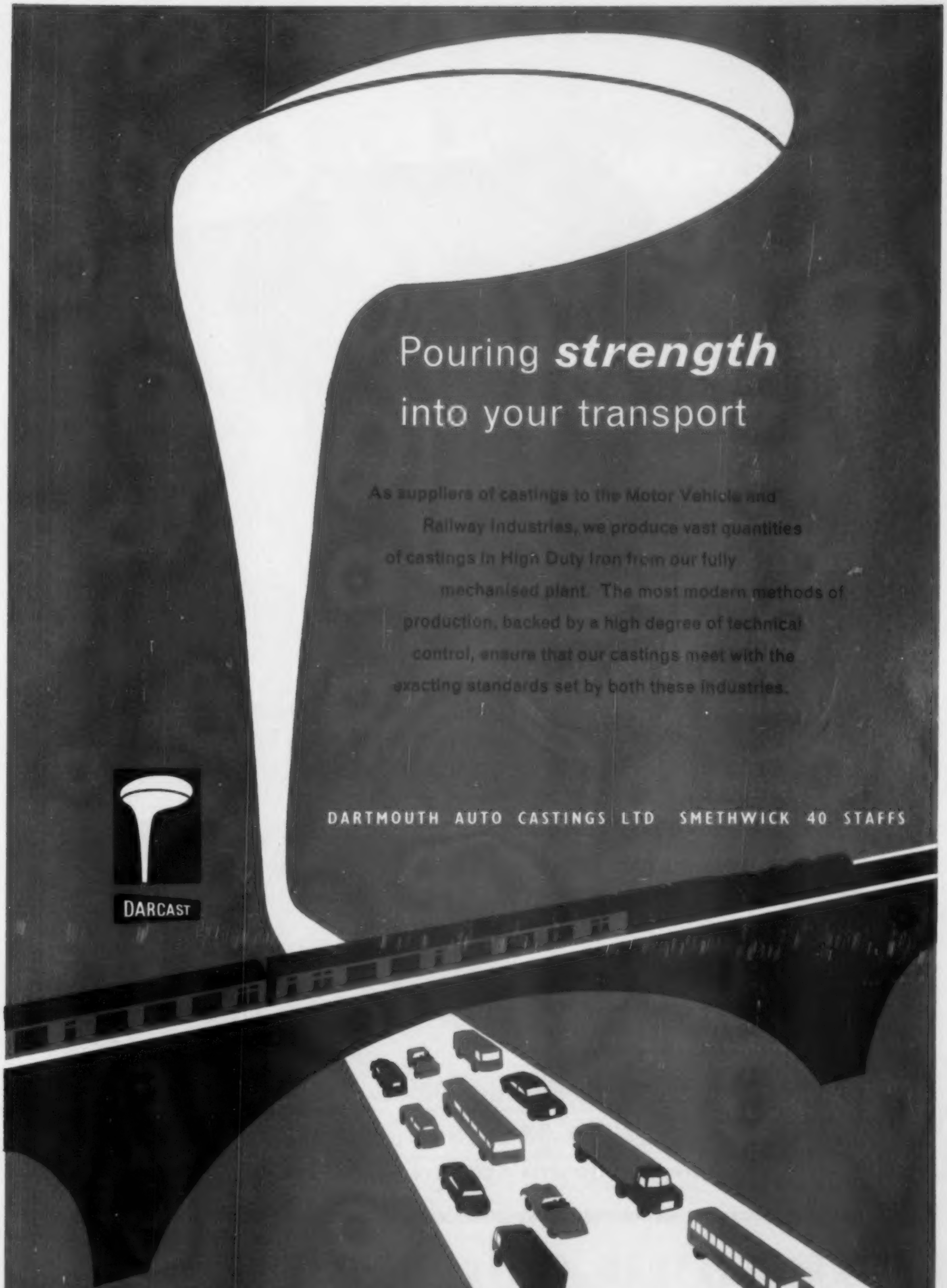
Single or double automatic cycle operation according to work.
Plunge cut and traverse grinding.
Feed accelerator reduces production time.
Exceptionally wide range of roughing and finishing feeds.
Diminishing rate of feed to zero with variable electric dwell at size position.
Adjustable oscillation for building-up finish on blind end and open short length bores.
Hand feed to dead stop for plain internal grinding.
Operational safety—provision for immediate interruption of automatic cycle for emergency run-out—enclosing of dogs and plungers eliminates 'finger traps'.
Can be equipped for combined hole and face grinding.

THE CHURCHILL MACHINE TOOL CO. LTD., BROADHEATH, near MANCHESTER
Telephone: Altrincham 3262 Telegrams: Churchale, Manchester
Export Sales Organisation: Associated British Machine Tool Makers Ltd., London, Branches and Agents
Home Selling Agents: Charles Churchill & Co. Ltd., Birmingham and Branches

PRECISION *plus* PRODUCTION

C661-1A

One good turn . .
Do your operators a good turn by providing them with *Osborn* cutting tools and you will do yourself a good turn by increasing efficiency and output.
An extensive range of lathe and planer tools in solid or butt-welded high-speed steel and *Osbornite* hard metal, together with toolholder bits, are produced throughout, from raw material to finished product, within the same organization.
OSBORN
MUSHET brands
S.O.B.V. &
OSBORNITE
TRIPLE MUSHET
SAMUEL OSBORN & CO., LIMITED.
CLYDE STEEL WORKS, SHEFFIELD.
Fine Steelmakers • Steelfounders • Engineers' Toolmakers


Pouring strength
into your transport
As suppliers of castings to the Motor Vehicle and
Railway Industries, we produce vast quantities
of castings in High Duty Iron from our fully
mechanised plant. The most modern methods of
production, backed by a high degree of technical
control, ensure that our castings meet with the
exacting standards set by both these industries.
DARCAST
DARTMOUTH AUTO CASTINGS LTD SMETHWICK 40 STAFFS

KE

THERMAL FATIGUE RESISTING STEELS

FOR DIE-CASTING, EXTRUSION, STAMPING & FORGING

HOT WORKING TOOL STEELS IN ULTRASONICALLY TESTED FORGINGS...

Our recent developments in this field are the result of careful research by our Technical Staff into the call for increased productivity. Should you wish for details of our specialities please notify us so that we may arrange for one of our Technical Representatives to call upon you.

KAYSER ELLISON & CO. LTD.

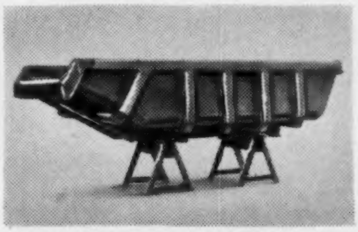
CARLISLE STEEL WORKS · SHEFFIELD · ESTABLISHED 1825

22 ton Euclid Rear Dump being loaded by large mechanical shovel.

CARRYING THE LOAD . . .

comes easily to the Euclid 22 ton Rear Dump Trucks specially built for "off-the-highway" heavy duty hauling of earth, rock, and ore on grade variation projects.

To endure this day-in, day-out punishment, heavy construction and first-class design are vital, together with welding that stands up to the continual stresses and strains. That is why the makers chose Actarc "VELVARC" Electrodes, whose superior characteristics are fully recognized by so many of the leading firms in the welding industry.



The first body produced at Euclid, Newhouse.

Euclid 22 ton Rear Dump

Photographs reproduced by kind permission of Euclid (Great Britain) Ltd., Newhouse, Scotland.

"VELVARC" is a general purpose Electrode giving a smooth velvety arc, self-removing slag with excellent mechanical properties. A wide range of other Actarc Electrodes is available. There are lime-ferritic electrodes for welding medium carbon and low-alloyed steel; mild steel electrodes for general purpose work; special rods for vertical and over-head welding, and for deep penetration; electrodes for fast horizontal production work; electrodes for boiler plate; non-ferrous electrodes for aluminium, nickel and bronze. Write, telling us of your specific requirements to:

ACTARC WELDING ELECTRODES

ARC MANUFACTURING COMPANY LIMITED

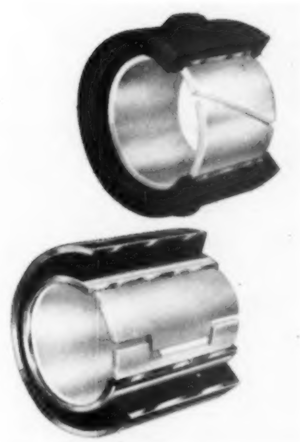
ACTARC WORKS, NITSHILL, GLASGOW, S.W.3 — *Telephone: Barrhead 2293*

London Office and Works: 15 HERCIES ROAD, NORTH HILLINGDON, MIDDLESEX

E 4

L26

AUTOMOBILE ENGINEER



THE UPPER OF THESE TWO NYLASTIC BEARINGS IS HOUSED IN A RUBBER SLEEVE, WHILE THE LOWER ONE IS IN A CLAYFLEX FLEXIBLE BUSH

Published the second Wednesday in every month
by ILIFFE & SONS LIMITED
Dorset House, Stamford Street, London, S.E.1
Telephone · Waterloo 3333 *(60 lines)*
Telegrams · Sliderule, Sedist London
The annual subscription inland and overseas is £3 0s 0d *including the special number*
Canada and U.S.A. $8.50

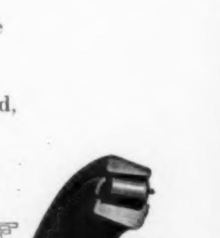
BRANCH OFFICES
Coventry · 8-10 *Corporation Street*
Telephone · Coventry 25210
Birmingham · King Edward House, New Street, 2
Telephone · Midland 7191
Manchester · 260 *Deansgate,* 3
Telephone · Blackfriars 4412 *and Deansgate* 3595
Glasgow · 26B *Renfield Street,* C.2
Telephone · Central 1265

CONTENTS

VOLUME 48 NUMBER 7 JULY 1958

The Conquest of Friction

Leonardo da Vinci, that versatile genius of the Renaissance, studied friction in many of its aspects, being the first to distinguish between sliding and rolling friction. He also remarked upon the importance of polishing the surface of parts that move in a groove or around a pivot. Among his many manuscripts are sketches of several ingenious anti-friction bearings which he described as ' pivots of the highest perfection ' and mechanisms whose motion, once started, were declared to be ' wonderful and supernatural in duration '. Some of these are known to have had practical application. Hundreds of years later many more of da Vinci's theories were to find practical expression and further development with the foundation of SKF and the introduction of a unique range of anti-friction bearings for many different purposes.



THE SKEFKO BALL BEARING COMPANY LIMITED · LUTON · BEDS

SKF

G151

THE ONLY BRITISH MANUFACTURER OF ALL FOUR BASIC BEARING TYPES: BALL, CYLINDRICAL ROLLER, TAPER ROLLER AND SPHERICAL ROLLER

Editor *J. B. Duncan*

Editorial Staff *T. K. Garrett, A.M.I.Mech.E. A.F.R.Ae.S.(Associate Editor)*

F. C. Sheffield

DESIGN MATERIALS **AUTOMOBILE ENGINEER** PRODUCTION METHODS WORKS EQUIPMENT

Range of Vision

FOR a number of years, the sales departments of some of the American motor manufacturers have in a relatively small way made a feature of road safety in the advertisements concerning their products. More recently, as a result of outside pressure, the designers and commercial departments of these manufacturers have been concentrating much more on safety and are now tending to discontinue competition, at least in advertising, in respect of engine power outputs. In many instances, emphasis appears to have been placed on the avoidance of injury to the occupants in the event of a crash, but the prime consideration obviously should be the incorporation of features designed to avoid accidents.

A number of reports describing research in this field in America has been published. Although these reports undoubtedly are of value, they should not be accepted without question as applicable in road and traffic conditions in the United Kingdom. It would appear that a high proportion of the accidents that occur in America are nose-to-tail collisions, and a great deal of the research that has been undertaken there is relative to those conditions. In Great Britain, extremely valuable work is being carried out by the Road Research Laboratory, who are making careful and detailed investigations of accidents, with a view to collecting indisputable facts as to the causes and effects. Such investigations, of course, take a long time to complete, and it would be premature to jump to conclusions until authentic information is available.

Investigations completed so far, however, have indicated that range of vision is undoubtedly a significant factor with regard to road safety. A useful contribution can be made by reducing the effective widths of the windscreen pillars and of the rear quarter panels above the waist. In general, the total effective width of the windscreen pillar, including the adjacent glazing frame of the triangular ventilating panel in either the open or closed positions, should be no more than about 2 in. The effective width is, of course, defined as the projection of the pillar on to a plane perpendicular to a horizontal line joining the centre of the pillar section and the point mid-way between the eyes of the driver.

Objections to the reduction of pillar sections are sometimes raised, on the ground that the roof will collapse in the event of the car's rolling over in a crash. However, in view of the infrequency of the occurrence of cars riding on their roofs, it is questionable as to whether they should be specifically designed so to do. Certainly, few people would be so bold as to suggest that the roof supports of British cars need to be any stronger than they are currently, since the evidence at present available seems to indicate that on the rare occasions when cars roll over, the roof does not normally collapse. However, this factor might have to be considered in relation to four-door, hardtop vehicles in which the pillar between the two doors on each side does not extend the whole distance between the sill and the cant rail, and in which, also, unusually slender pillars are employed at both the front and rear quarters.

So far as the view in the mirror is concerned, despite the fact that the driver's head inevitably blocks an appreciable portion of the rearward range of vision, the wider the rear light the better. A factor that is sometimes overlooked is that a wide rear light contributes in no small measure to road safety by giving the driver of a following car a clear view of the road ahead, through the vehicle in front of him: thus he is able to see obstructions that would otherwise be invisible to him, and to anticipate the action that the driver in front will take.

The position of rear-view mirrors is obviously important. Nevertheless, it is not always accorded due consideration during the initial design of the vehicle. In this context, so far as private cars are concerned, the upper edge of the rear light should be at least 4 in higher than the level of the driver's eye, and the mirror should be positioned in such a way as to give a horizontal line of sight to the rear and a field of view extending above the horizontal. If the top edge of the rear light is not high enough, it might only be possible to meet these requirements by placing the mirror several inches below the head rail, or even on the dash facia. In these circumstances, the mirror is liable to obstruct the range of vision in the forward direction. If it is mounted on the facia, the range of vision to the rear may be obstructed by the shoulders, as well as the heads, of the driver and passengers. Externally-mounted mirrors are undoubtedly useful in indicating to a driver that he is about to be overtaken, when the overtaking vehicle is outside the range covered by the internally mounted mirror.

Frequently, the significance of the range of vision downwards over the front of the vehicle is not fully appreciated. There have been many instances of drivers of vehicles pulling away from a kerb without having been able to see obstructions such as bicycles, or even children, on the ground immediately in front of their near-side wheels. In foggy conditions, the distance between the driver's eyes and the nearest visible points on the kerb and the road in front should be as short as possible. In fact, the ease with which motor cyclists can see in fog, relative to the drivers of four-wheeled vehicles, possibly is not entirely accounted for by the windscreen between the driver and the road.

ANTI-SKID DEVICE

Preliminary Report on a Dunlop Development that is a Significant Contribution to Road Safety

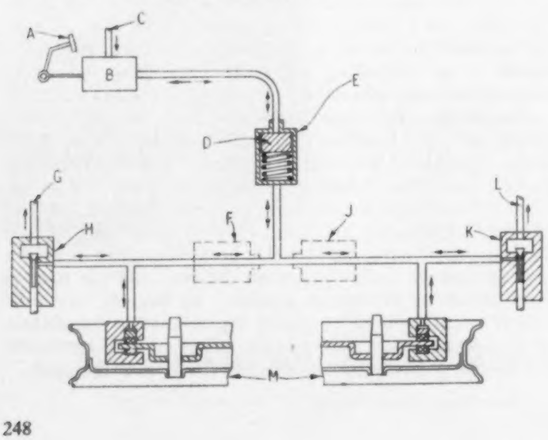
FOR several years, an anti-skid device suitable for application to both private cars and commercial vehicles has been under development by the Aviation Division of the Dunlop Rubber Co., Ltd., of Foleshill, Coventry. The new unit has been evolved from the Dunlop Maxaret anti-skid device for aircraft, which was first introduced in 1952. Production of the Maxaret unit in this country has run well into five figures, and it is now being made under licence in three other countries, including the United States of America. The success achieved with this unit led to many enquiries being received with regard to the possibility of its adaptation for use on road vehicles. In fact, the early tests for automobile applications were made with aircraft type units.

The first tests of this kind were performed at the Government-sponsored Road Research Laboratory, where it was found that, under heavy braking conditions, the application of this unit gave a marked improvement in both controllability and directional stability. A film of some of the tests made by the Laboratory shows how a car fitted with a Maxaret anti-skid device on each wheel can be steered, to avoid a succession of obstacles, while the brakes are fully applied. Under these conditions, but without the unit, a car would slide completely out of control.

As a result of the considerable interest shown in the unit abroad, it was decided to design an anti-skid device specifically for road vehicles. This interest confirmed the opinion already held by Dunlop that, while there were already markets for such a unit in both the private car and heavy commercial vehicle fields, these markets would mainly be confined to the countries where very long journeys are common and where modern motor roads permit high cruising speeds to be maintained by all classes of vehicle. In Great Britain, it is expected that initially a market will be found mainly in the high-performance car class, and for application to public utility vehicles, such as police cars, fire engines and ambulances, which are occasionally called upon to travel at abnormally high speeds. However, it is possible that, even for commercial vehicles legally limited to 30 m.p.h., the increased safety that will be obtained with anti-skid devices fitted will be reflected in reduced insurance premiums. In this event, of course, a ready market would develop. The device would appear to be of particular value for application to semi-trailers to help to prevent jack-knifing.

Fig. 2. Diagrammatic layout of the brake system for the Maxaret unit

A brake pedal; B brake control valve; C from pressure supply; D movable restrictor, to give unrestricted initial flow to fill the brake cylinder; E modulator; F and J alternative positions of the modulator, to give individual control of the brakes; G and L exhaust lines to tank; H and K anti-skid units; M brakes



Principle of operation

The principle of operation is the same for both the aircraft and the road anti-skid units, and is very simple. A small flywheel is rotated by each braked wheel: as the road wheel is decelerated, so also is the flywheel. A spring is incorporated in the drive between the road wheel and flywheel, so that if the rate of deceleration of the road wheel is greater than that which would be experienced during normal braking, that is, if the wheels tend to lock, the spring deflects and allows relative movement to take place between the flywheel and its drive. This relative movement actuates a small hydraulic valve to relieve the brake pressure, until the rate of deceleration is reduced and the flywheel is returned by the spring to its normal position relative to the drive. If the wheels do not tend to approach the point of locking when the brakes are applied, the rate of deceleration is not sufficient for the inertia of the flywheel to deflect the spring.

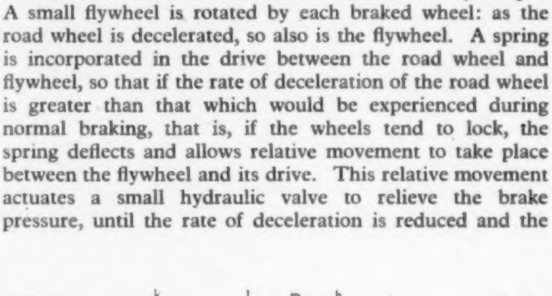


Fig. 1. General arrangement of the Maxaret anti-skid device that has been developed for road vehicles. This unit weighs only 2 lb 5 oz

A sectional arrangement of the anti-skid device currently undergoing development which, like all the other components of the system, is the subject of patent applications, is shown in Fig. 1, and its installation and drive arrangements in front and rear wheel applications are shown in Figs. 4 and 5. The main spindle *a* of the unit is driven at its right-hand end, as shown in Fig. 1. This spindle is carried in a ball bearing *b* in a housing at the right-hand end of the casing and a needle roller bearing *c* at the other end. Between these bearings, there are two more, on which is mounted the cam sleeve *d*. One is the ball bearing *e* in the right-hand end of the sleeve, and the other is the needle roller bearing *f* in the left-hand end. The drive is transmitted from the main spindle to the cam-sleeve by the cross pin *g*. On each end of this cross pin there is a ball bearing *h*, which seats in a V-shape cam-slot *j*. The cross pin is carried in diametrically opposite slots milled longitudinally in the main spindle, so that it can move freely in a direction

parallel to the axis of the spindle, but it cannot rotate about this axis. In fact, the cross pin is pushed by the main spring *k* towards the right-hand end of the slots in the spindle, so that the bearings at its ends normally seat in the base of the V-shape cam-slot in the sleeve.

If the road wheel tends to lock, and its rate of deceleration, therefore, is high, the two ball bearings ride up the arms of the V-shape slots, so the cross pin moves axially relative to the drive spindle, against the compression in the main spring *k*. In these circumstances, the drive is positively transmitted from the spindle, through abutment faces to the left of the cross pin, to the sleeve. The relative movement between the spindle and the cam sleeve is restricted to 30 deg by the amount of the clearance between the abutment faces. When the rate of deceleration is reduced again, the main spring returns the cross pin and ball bearing assembly so that the bearings once again seat in the base of the V; then the drive is transmitted through them instead of through the abutment faces. From the cam sleeve, the drive is transmitted through a clutch *l* to the flywheel *m*. The clutch mechanism allows a small amount of slip to

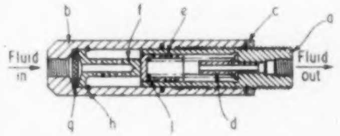


Fig. 3. Cross section of the modulator unit, which weighs approximately 1 lb

take place between the flywheel and cam sleeves during violent deceleration of the unit, such as would occur if the road wheels were to lock momentarily owing to a sudden drop in the coefficient of friction between tyre and ground.

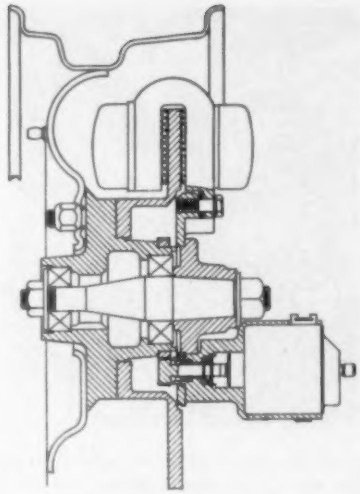
Axial movement of the cross pin, which amounts to approximately 0·060 in to the left, from the normal position, first closes a small clearance between the foot of the valve *n* and the spring support *p*, and then lifts the poppet valve off its seat. The precompression in the main spring determines the angular deceleration of the flywheel at which the valve is opened. When the poppet valve is lifted off its seat, brake fluid flows through connection *q*, past the valve and out through connection *r*, to relieve the brake pressure until the deceleration is reduced and the pin catches up with the sleeve, thus seating the valve again.

Carried in the hollow spindle *a* are the guide *s*, for the cross pin, and the valve assembly. Of these two components, the guide rotates with the spindle, but the valve assembly does not. The main component of the valve assembly is the seat *t*, which has two synthetic rubber seals round its periphery, to prevent fluid from bypassing the valves, by leaking through the clearance between the valve seat barrel and its housing in the end cover of the unit. There are two springs in the valve assembly; one is the main spring *k*, already mentioned, and the other is the valve return spring *u*. The arrangement of this assembly is clearly shown in the diagram.

Modulator

A diagrammatic illustration of the layout of the hydraulic system is shown in Fig. 2. Fluid from the brake control valve, which takes the place of the conventional master cylinder, flows through a modulator unit to the brakes. An anti-skid unit is connected to a T-junction in the pipeline between the modulator and each brake. The function of the modulator is to allow the brake fluid to flow initially without restriction, to take up the brake clearances, so that the brakes are applied without any time lag. Once they are applied, further flow through the modulator is restricted so that when the valve in the anti-skid unit opens, to relieve the pressure in the line to the brake, the outgoing flow of fluid through the relief valve is much greater than

Fig. 4. Illustration showing the experimental installation of the anti-skid unit on the front wheel assembly of a large car



the incoming flow through the modulator. This is to ensure that the brake pressure is rapidly released to correct the skid. When the anti-skid valve closes again, the rate of pressure rise in the pipeline to the brakes is determined by the degree of restriction in the modulator.

The arrangement of the modulator unit currently being developed is shown in Fig. 3. For the development work, it was necessary to incorporate an adjustment device, by means of which the volume of the free flow fluid permitted by the modulator can be matched to the total hydraulic volume required for the operation of the brakes. However, production modulator units are not expected to be adjustable, and, therefore, will be more simple in construction. The adjustment is provided for by screwing the inner sleeve *a* into the housing *b*, and the setting is fixed by the lock nut *c*. This adjustment of the inner sleeve relative to the housing regulates the position of the stop *d*, which limits the stroke of the restrictor piston *e*. Movement of the piston in the opposite direction is limited by the stop *f* which, together with the filter *g*, is retained by the snap ring *h*, in the end of the housing.

When the brakes are applied, the fluid passes through the filter and out through the radial holes in the stop *f*. It then forces the piston *e*, against the action of its return spring *j*, towards the stop *d*. As the piston moves, it displaces fluid through the axial hole in the stop *b* and out into the pipeline to the brakes. When the brakes are fully applied, the

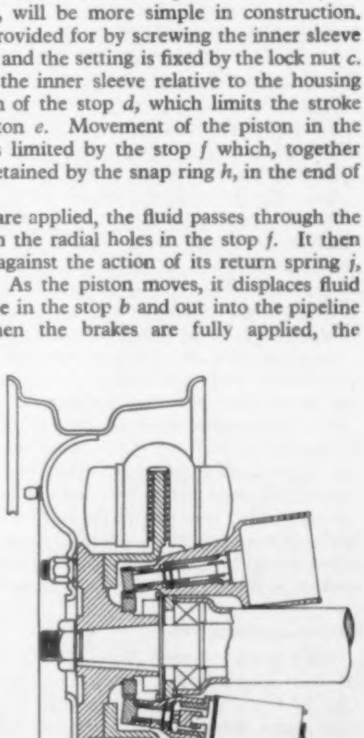


Fig. 5. Installation of the anti-skid unit, and a D.C. generator for signalling road speed, on a rear wheel assembly of the car used for the braking tests

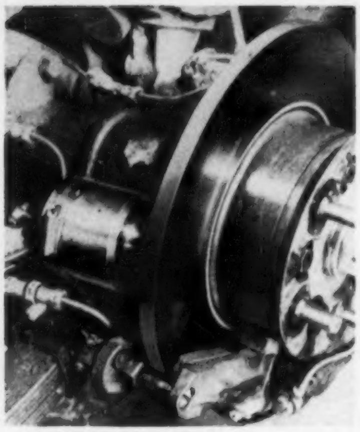


Fig. 6. This illustration shows the Maxaret unit installed on a disc type brake on a rear suspension assembly of a car

restrictor piston is on the stop *d*, so that when the valve in the anti-skid device is opened and closed, replacement of the brake fluid released can only take place through the spiral restrictor groove round the periphery of the piston *e*.

Installation

For the purpose of the development tests, the anti-skid device was installed on the front and rear wheel assemblies of a Mark VII Jaguar, as shown in Figs. 4 and 5. There is also a half-tone illustration of the front wheel installation, Fig. 6. At both the front and the rear, a gear ring is pressed on to the road wheel hub, and the anti-skid unit is driven by a pinion giving a ratio of 4·57:1. This drives the anti-skid unit at a speed of 5,500 r.p.m. when the road speed is 100 m.p.h. The rolling radius of the tyre is approximately 14 in. To prevent dirt from fouling the gears, they are protected by a close-fitting cover.

In Fig. 5, which shows the rear wheel assemblies, the installation of the tachometer generators can also be seen. These generators are driven in the same way as the anti-skid units, and are used to record variations in wheel speed during the test. Lack of space prevented the fitting of tachometer generators to the front wheels.

It is expected that the anti-skid units in their final form will be driven by a ring gear attached to, or forming part of, the brake disc. In these circumstances, the anti-skid unit could be supplied as an integral part of the body of the disc brake. This would obviate all installation problems, so far as the vehicle constructor is concerned. Suggested schemes with these features are shown in Figs. 7, 8 and 9.

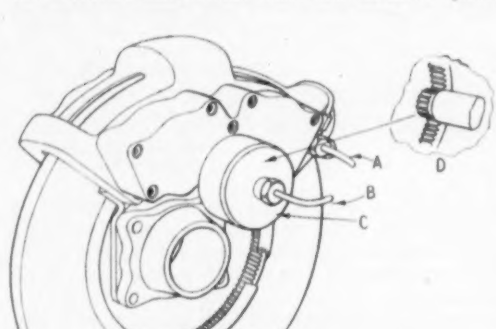
Although all the illustrations show the unit used in conjunction with disc type brakes, it can also be applied to the drum type, provided the shoe layout is not such as to give a mechanical servo action. In other words, it can be incorporated with the two-trailing-shoe type layout, but not when leading shoes are employed. The reason why it cannot be used in conjunction with a mechanical servo layout is that, since the brake torque is not proportional to brake pressure in these systems, the fluctuations in braking effort would be too violent, as the pressure in the hydraulic system is alternately relieved and reapplied.

Brake operating system

Since similar systems have already been used on aircraft, the test installation was built up of aircraft components. The layout is shown diagrammatically in Fig. 10. A gear type pump, driven at approximately 0·6 times engine speed, draws fluid from a tank mounted above the engine, and supplies it through a filter to an automatic cut-out. This cut-out is set to supply fluid to accumulators until the pressure rises to 1,700 lb/in^2, when it diverts the fluid from the pump into an idling circuit. When the pressure in the accumulators falls to 1,500 lb/in^2, the unit again is brought into operation to raise it to the cut-out pressure. Two 100 in^3 accumulators are used, and they are charged initially with air at 1,000 lb/in^2.

The pressure applied to the brakes is regulated by an aircraft type brake control valve, actuated by a linkage connected to the brake pedal. This control valve is of the fully-powered type, that is, a continuous supply of fluid under pressure is available to re-apply the brakes after the anti-skid unit has released them. Such an arrangement is necessary, of course, because a conventional master-cylinder type of brake control unit does not meet the requirements of the anti-skid unit with regard to continuous supply of fluid.

From the brake control valve, the fluid is fed through two modulator valves; from one of these it goes to the front brakes and from the other to the rear brakes. Beyond each modulator, the pipe connection forks to each side of the car. This piping arrangement couples the brakes in pairs, so that a skid on either front wheel will cause the pressure to



A *supply line;* **B** *return to tank;* **C** *anti-skid unit;* **D** *drive to anti-skid unit*

Fig. 8. A diagrammatic illustration of a proposed scheme for incorporating the Maxaret unit integrally with a twin-spot type disc brake

be relieved in both front brakes. The same applies to the rear wheels.

As has already been mentioned, the connection to each anti-skid unit forms the leg of a T-junction in the brake line to the wheel. A single pipe from the front and another from the rear return to the tank the fluid exhausted by the units during skid correction. Fluid only flows in these two return pipes when a skid is being corrected: when the driver releases the brakes, the fluid exhausted from the wheel cylinders is returned through the brake control valve to the tank.

For development purposes, a solenoid valve was fitted in the supply line to each anti-skid unit. Normally, both these valves are left open, and they can be controlled by means of two switches mounted on the dash. When the valves are closed, the anti-skid units are cut out of action, so that comparisons can be made between the performance with the anti-skid control and that with the normal control system in operation. A proposed hydraulic circuit, incorporating a vacuum operated, hydraulic power pack, is illustrated in Fig. 11. This system will be discussed later, under the heading "Future developments".

Instrumentation

Since accurate records of decelerations obtained were required, an instrument of the U-tube type, recording peak deceleration, the accuracy of which can be affected by pitching, was of no use. Moreover, an instrument of this

type cannot be used to record average deceleration, for calculating the stopping distance. Therefore, a 12-channel mirror galvanometer type recorder was employed.

The recorder, which was mounted in the boot, is shown in Fig. 12. This instrument records on 15 cm wide, photographic paper. It marks a 0·1 second time base at a paper-speed of approximately 1·0 inches per second. Five pressure pick-ups of the strain gauge type were fitted in the hydraulic system, to signal the pressure at each brake and the pressure selected by the driver. The speed of each rear wheel is signalled by the tachometer generator, already mentioned. Accelerations in the fore-and-aft plane are signalled by an instrument mounted on the floor of the boot: this unit is marked A in Fig. 12.

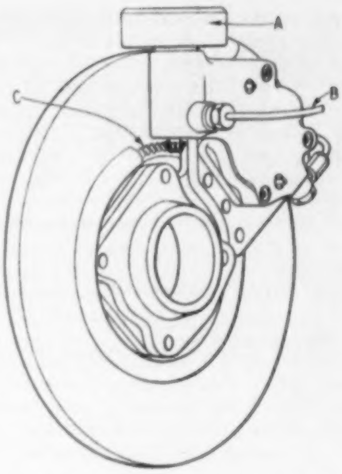


Fig. 7. Anti-skid unit arranged integrally with a disc brake of the single-spot type

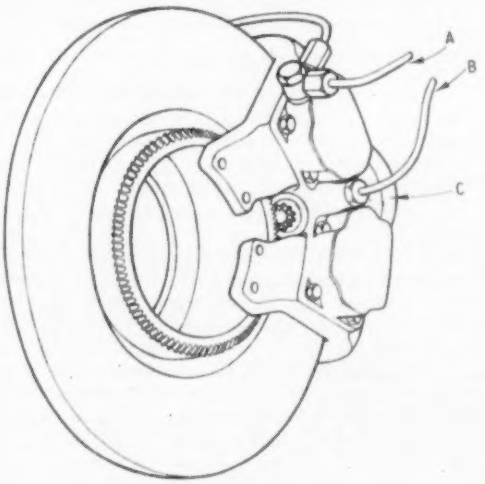
A *anti-skid unit;* **B** *return line to tank;* **C** *drive to anti-skid-unit*

Test procedure

Tests were made with treadless tyres running on a specially laid skid track flooded with water, and treaded tyres running on dry coarse-textured surfaces. These conditions gave coefficients of friction between the tyres and the road, ranging between the values of 0·1 and 1·0. Speeds on the skid track were limited by the length of the track available to approximately 45 m.p.h. However, tests under conditions giving coefficients of friction in the upper part of the range were made on a disused aerodrome runway of coarse-grained asphalt, where speeds of up to 95 m.p.h.—the maximum obtainable with the car—could be attained. The surface of the runway could, of course, be wetted as required.

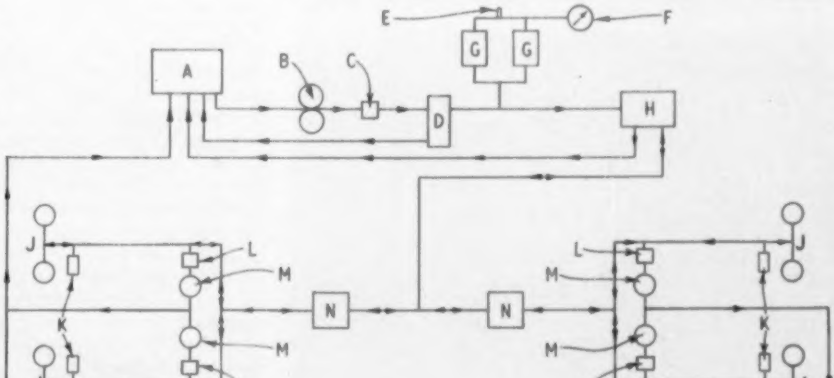
Normally, the brakes were applied fully when making the tests, the brake pressure obtained being regulated by means of an adjustable stop that limited the pedal movement. In each case, the car was first braked with the anti-skid units in operation, and then the test was repeated with the units cut out by means of the switches on the dash. This procedure enabled a direct comparison to be made between anti-skid and normal braking conditions, while the conditions under which the comparative tests were made were virtually identical.

In most cases, the application of the brakes without the anti-skid units in operation caused the car to slide to a standstill with all four wheels locked. The brake pressure was about 1,000 lb/in^2, and the distribution of braking effort between the front and rear wheels was in the ratio of 60:40. During the earlier tests, calculations of average deceleration were based on the initial velocity and the stopping time, both of which are recorded accurately by the instrumentation system. More recently, a road marker, operated by an electrically fired explosive charge, was employed. This marker is triggered by means of a switch on the brake pedal. With this installation, the actual stopping distance can be measured accurately, so the average deceleration calculations can be based on the more significant criteria, of initial speed and stopping distance. Instantaneous decelerations are recorded by the 12-channel apparatus. The experimental car, equipped with the anti-



A *supply line;* **B** *return to tank;* **C** *anti-skid unit*

Fig. 9. Above: An alternative method of mounting the Maxaret anti-skid unit integrally with a twin-spot type disc brake

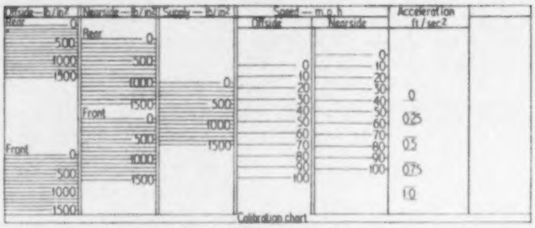


A *supply tank;* **B** *Plessey pump;* **C** *filter;* **D** *Lockheed automatic cut-out;* **E** *charging connection;* **F** *gauge;* **G** *accumulators;* **H** *Dunlop Mk III brake control;* **J** *brakes;* **K** *pressure pick-ups;* **L** *Electromatic valves;* **M** *anti-skid units;* **N** *modulators*

Fig. 10. An experimental hydraulic circuit for a car brake system incorporating the anti-skid units

RECORDS TAKEN DURING THE TEST RUNS

On the left is the calibration chart applicable to all the diagrams, and the key to the abbreviations used is on the left of Fig. 15, run 1



Calibration chart

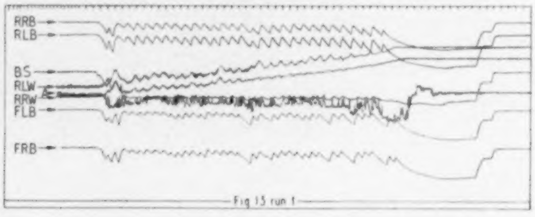


Fig 13 run 1

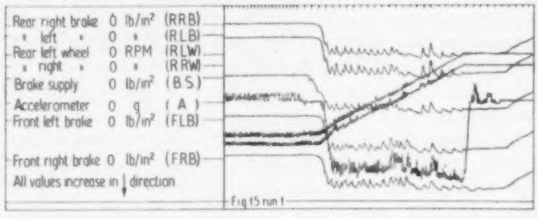


Fig 15 run 1

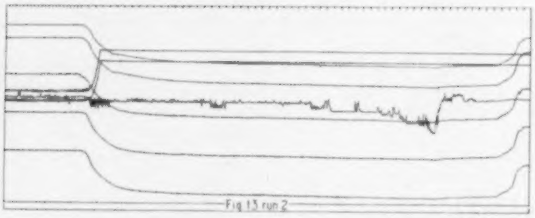


Fig 13 run 2

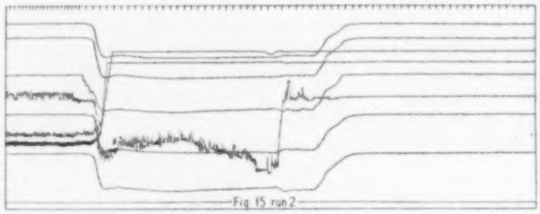


Fig 15 run 2

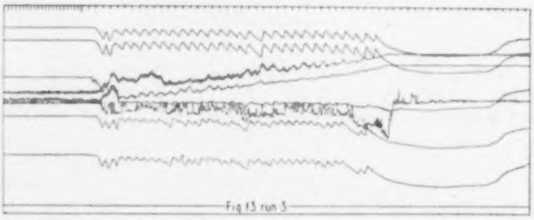


Fig 13 run 3

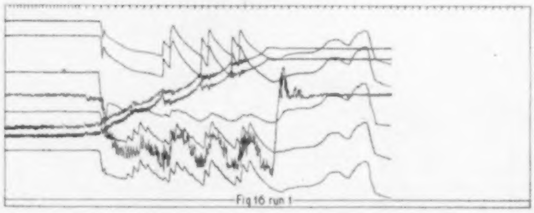


Fig 16 run 1

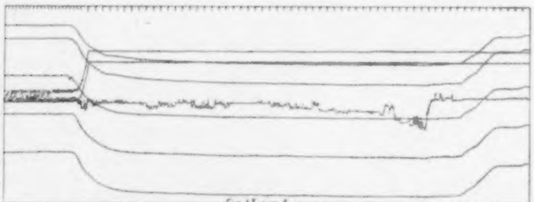


Fig 13 run 4

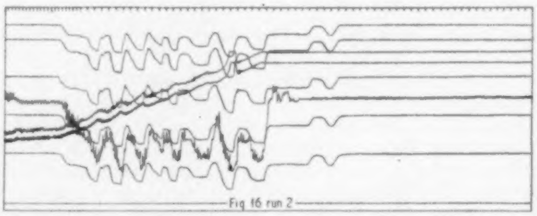


Fig 16 run 2

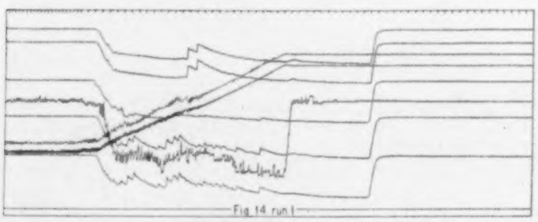


Fig 14 run 1

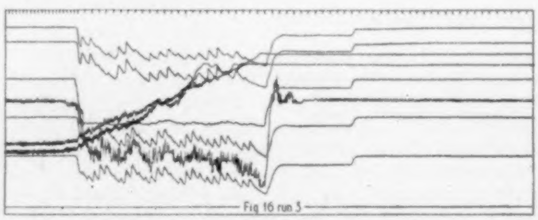


Fig 16 run 3

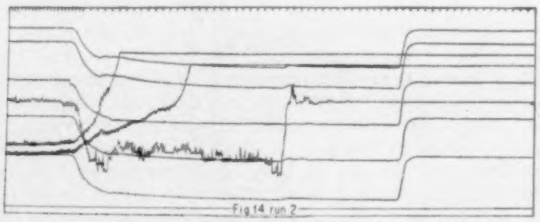


Fig 14 run 2

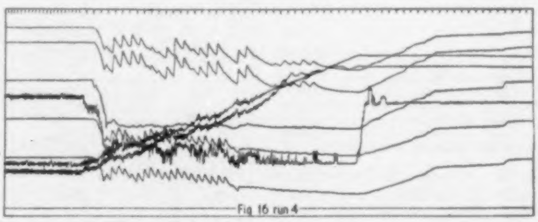
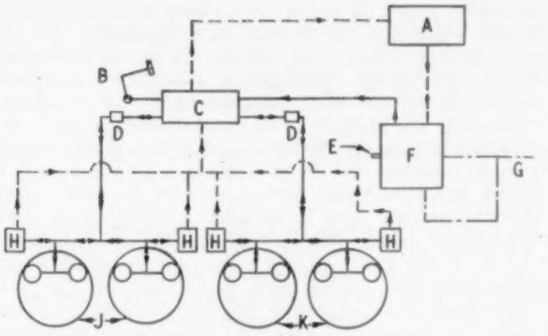


Fig 16 run 4



——— supply; - - - low pressure; — — — — — — vacuum

A supply tank; **B** brake pedal; **C** power valve; **D** modulators; **E** air charging valve for accumulators; **F** hydraulic power pack, vacuum operated; **G** vacuum line; **H** anti-skid units; **J** front brakes; **K** rear brakes

Fig. 11. A proposed hydraulic circuit, incorporating a vacuum operated hydraulic power pack, for use with the Dunlop Maxaret unit

skid device was tested by *Automobile Engineer* staff on the skid track at Fort Dunlop and on the aerodrome. For the purpose of these tests, the car was fitted with conventional tyres and, therefore, was in a normal condition, except inasmuch as it was loaded with test equipment in the boot. So far as performance was concerned, the Maxaret unit did all that has been claimed for it. Undoubtedly, the most important feature is that even when the brakes are fully applied by the driver, the vehicle can be steered easily around obstacles on a slippery surface.

TEST RESULTS

Smooth tyres on flooded skid track

Records made during tests on various surfaces are reproduced in Figs. 13, 14, 15 and 16. The first of these is a record made during a test with smooth tyres fitted to a car and running on the flooded surface of a skid track. This condition is approximately the same as that of a car with normal tyres running on ice. Runs 1 and 3 were made with the anti-skid units in operation and runs 2 and 4 with them switched off. In each case, the brake pressure selected was 1,000 lb/in^2, although, as can be seen from the diagrams, the continuous cycling of the anti-skid units causes a considerable drop in the supply pressure. This drop can be attributed to shortcomings of the type of control valve used and would not occur with the valve that has since been designed specifically for this type of application. On all runs, the brakes were applied when the vehicle was moving at 35 m.p.h., and the following stopping distances were measured: run 1, 207 ft; run 2, 257 ft; run 3, 202 ft; run 4, 269 ft.

An increase in coefficient of friction as the car speed decreases is reflected in the wheel speed trace obtained when the anti-skid unit is in operation. Unfortunately, when the wheels are locked, there is no indication of the car speed; however, from observation, the difference in deceleration rates, as between the runs with the anti-skid units in and out of operation, was most marked during the first third of the braking run: when the wheels were locked, the car lost very little speed during this period, whereas with the anti-skid units in operation, deceleration was apparent immediately the brakes were applied. This, of course, is reflected in the difference in stopping distances, and it also shows up on the deceleration trace.

In some other tests, the driver was asked to attempt to regulate the pressure, simply by manipulating the brake pedal, to a value that would give satisfactory deceleration without the wheels locking. However, there was so little difference in the feel, as between the wheels in the locked and the unlocked conditions, that this was impracticable.

Cornering tests with treaded tyres

Tests were also carried out on the flooded skid track, but with treaded tyres fitted, to determine the maximum speed at which the car would take a curve of 150 ft radius without sliding outwards. With the brakes off, this maximum speed was 40 m.p.h.; if the brakes were applied at approximately 38 m.p.h., it was found that, without the anti-skid units in operation, the rear end of the car slid outwards so that the vehicle took up a new position, with its nose pointing in a direction nearer to the centre of the radius. This sliding invariably took place as soon as the brakes were applied, and was too quick for the driver to correct it before the car struck markers that originally were about 3 ft from the car on the inside of the radius.

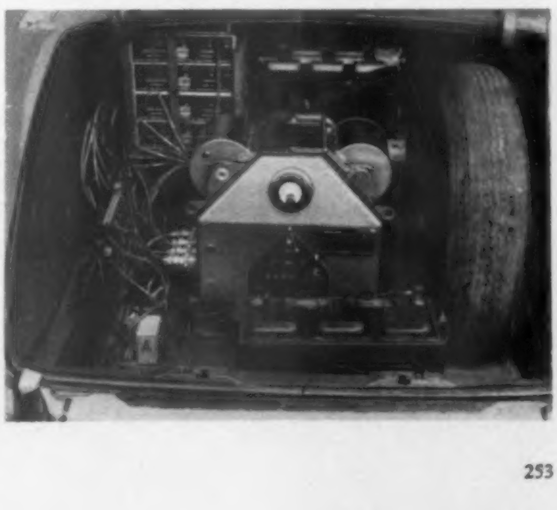
With the anti-skid units in operation under the same conditions, there was a slight outward twitch of the back of the car immediately the brakes were applied, as the first operation of the anti-skid units took place. However, this caused no appreciable deviation in the course of the car and was in no way an embarrassment to the driver.

Normal treaded tyres on asphalt

A record taken with normal tyres fitted to the car running on a wet surface on coarse-grained asphalt is shown in Fig. 14. This surface was of a porous nature and would not hold water at an appreciable depth on it, as would the skid track. As before, the first run was with the anti-skid units in operation and the second without. In each case, the brakes were applied at 76 m.p.h., and the stopping distances measured by using the explosive road markers. The distances were 310 ft and 340 ft respectively. From these figures, the mean deceleration values can be estimated, using the formula $2(s-ut)/t^2=a$, where s is the stopping distance, u the initial velocity, and t the time. The values obtained were 19·7 ft/sec^2 and 18·2 ft/sec^2, which represent 0·612 g and 0·565 g respectively.

From the record, it can be seen that on run 2, without the anti-skid devices, the left-hand rear wheel locked 1·2 sec after the brakes were applied, and the right-hand rear wheel 3·2 sec after the application. The front wheels were observed to lock within 0·3 sec of the brake application, although this, of course, was not recorded. These figures suggest that the braking overload was far greater on the front wheels. This conclusion is borne out by the slight operation of the rear anti-skid units and the almost continual operation of the front ones. The reason why these conditions arose was that the rear wheels were more

Fig. 12. The boot of the test car, showing the 12-channel recorder, batteries and the instrument, marked A, for measuring deceleration



heavily loaded than normal, because of the weight of the test equipment that had to be carried in the boot.

Part-worn tyres on wet asphalt

The record shown in Fig. 15 was made with normal tyres on wet asphalt, but in this case the tyres were part-worn. Again, the first run was made with the anti-skid units in operation, and the second without. On the second run, all the wheels locked within 0·5 sec of brake application. Measured stopping distances are not available for these runs, which were made from speeds of 69 and 71 m.p.h. respectively. Calculation of deceleration, based on the initial velocity and time only, for the first run, gives a figure of 0·765 g; this is the highest recorded so far during the tests on any combination of tyre and surface. Comparison of the decelerometer traces of the two runs clearly shows the low deceleration obtained with all wheels locked, until the speed dropped to about 20 m.p.h. On the other hand, with all wheels fully braked but not locked, that is, under control of the anti-skid units, the rate of deceleration is high right from the point of brake application.

Treaded tyres on dry asphalt

A record taken with treaded tyres on the same asphalt surface, but in this case dry, is shown in Fig. 16. The first and second runs were both made at speeds of 68 to 70 m.p.h. In the first, the anti-skid units were in operation and full brake pressure was applied by the driver. However, in the second, the anti-skid units were switched off, but the driver relieved the brake pressure each time he felt that a skid was developing: this was done in an attempt to obtain maximum deceleration, while still maintaining directional control of the car. It can be seen that a high degree of skill was exercised by the driver, since the pressure was not at any time fully relieved. A relatively high rate of deceleration was maintained during the whole of the stop, and the wheels only locked for a short time, towards the end. Stopping distances of 232 ft and 248 ft were measured. These distances show that, despite the high degree of the skill of the driver, the use of the anti-skid units still gave a marked reduction in the stopping distance.

The third and fourth runs were both made with the anti-skid units in operation, the brakes being applied at 76 and 92 m.p.h. respectively. Stopping distances of 302 and 470 ft were measured. This represents decelerations of 19·7 ft/sec^2 and 19·3 ft/sec^2, or 0·612 g and 0·6 g. It can be seen that at the end of the third stop, the wheels locked for a short time on the right-hand rear wheel. In the system used on the test car, the brakes were hydraulically interconnected and, therefore, the pressure in both rear brakes must have been the same at all times; and since there is no evidence of locking of the left-hand rear wheel, the locking of the right-hand one was attributed to slow release of that particular brake.

Consideration of test results

As might be expected, the lower is the tyre-to-ground friction coefficient, the greater the advantage, so far as stopping distance is concerned, of the anti-skid system over the normal arrangement. However, at speeds of more than 60 m.p.h., the advantage of the anti-skid unit is very marked for all conditions of tyre-to-ground friction. It enables the driver, irrespective of his ability, to maintain maximum deceleration while still maintaining complete directional control over the car. This conclusion is fully confirmed whenever the car, without anti-skid units, is driven by a person of normal driving skill: invariably the driver releases the brakes as soon as the car makes the slightest deviation from its normal path, although in many instances the wheels have not fully locked. This reaction on the part of the driver would, of course, be accentuated under normal driving conditions, for it must be remembered that the high-speed test work was carried out on an aerodrome runway, 150 ft wide. Independent tests will be carried out shortly on the car driven under normal road conditions. These tests will be made by the Road Research Laboratory, who can arrange for the closing of public roads for this purpose, and they will be the subject of an official report.

One important factor demonstrated by the tests is that with the anti-skid units in operation it is possible to make considerably greater than normal braking power available to the driver, so in the early part of the development programme the braking capacity on the test car was increased to take advantage of this fact. In addition to the obvious advantage, of better deceleration under good tyre-to-ground conditions, the provision of more powerful brakes could be of considerable value in that, used in conjunction with the anti-skid units, they would allow for variations in brake performance due to fade and other factors.

Tests so far have been confined to a private car loaded to a constant weight. However, it is expected that the anti-skid system will prove to be of greater advantage on commercial vehicles. This is because the very high variations between the laden and unladen weights of these vehicles have for so long prevented the efficient use of the brakes, as has been pointed out by Starkes and Lister in their paper entitled "Experimental Investigations on the Braking Performance of Motor Vehicles", read before the Institution of Mechanical Engineers in 1954. An anti-skid system would not only cater for variations in the normal tyre-to-ground friction, but also would automatically adjust the braking torque at each wheel, to suit the weight carried by that wheel. The extra safety that such a system could bring to the operation of heavy vehicles, particularly the articulated, or semi-trailer type, could easily result in this system being considered essential in countries where vehicles of this type are not subject to speed restriction.

Future developments

As mentioned earlier in this report, the anti-skid unit is not suitable for use with the normal type of hydraulic system at present employed on road vehicles, since this system cannot be made to compensate for fluid exhausted during anti-skid operation. Therefore, the Dunlop Rubber Co., Ltd., is carrying out a design study on a new type of power brake system suitable for use on all types of road vehicles. This system, which is shown diagrammatically in Fig. 11, not only meets these requirements, but also has several advantages over existing arrangements.

Fluid is fed under pressure to the power valve, which under normal operating conditions regulates the pressure passed on to the brakes. This pressure supplied to the brakes is regulated in proportion to the load applied to the brake pedal, as distinct from the travel of the pedal: in fact, with this system, the pedal has virtually zero travel. The power valve maintains the selected pressure, by replacing the fluid exhausted during anti-skid operation.

Under abnormal conditions of operation, that is, in the event of failure of the power supply, the power valve continues to operate normally with fluid stored under pressure in the power pack. However, the stored energy is only adequate for a limited number of brake applications: when it is exhausted, extra travel of the brake pedal brings into action a piston operating on the conventional master cylinder principle: in other words, the system reverts to conventional operation. The extra travel of the pedal can be used to cut off the exhaust flow from the anti-skid units, so that they do not operate under these abnormal conditions. This ensures that the fluid is not exhausted from the brakes when they are actuated by the master cylinder.

FLEXIBLE BEARINGS

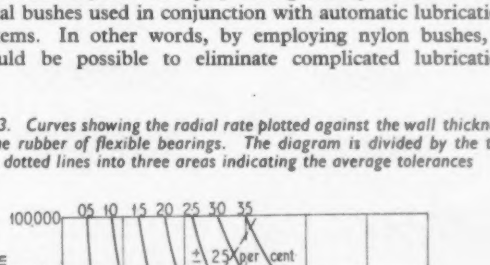
Some of the Fundamental Principles Involved in the Employment of Rubber and Nylon Bushes

SINCE there are so many advantages associated with the employment of flexible bushes, it is not surprising that this type of bearing has become widely favoured for application to many of the mechanisms employed in motor vehicles. Hitherto, the flexible element of these bushes has been almost universally of rubber, but for certain applications, nylon is now favoured. The merits of rubber bushes can be summarized as follows. Provided suitable bushes of good design are employed, they should last the life of the vehicle. Since they do not wear, there is no danger of rattles developing in service. By virtue of their flexibility, they are able to absorb high frequency vibrations. Rubber bushes also eliminate lubrication points and do not require any attention during service.

Nylon bushes are sometimes employed where the loading is too high for rubber ones of a similar size, and where space restrictions prohibit the use of larger bushes. There is a major difference between the principle of operation of nylon bushes and that of rubber bushes: whereas there is no relative sliding between the rubber bush and its pin or housing, the nylon bush is employed in much the same way as a metal one, that is, relative sliding motion takes place between the bush and either its pin or housing, or between both.

The merits of nylon bushes are as follows. Because of its resilience, this material is capable of absorbing a certain amount of high frequency vibration, although not as much as rubber. The resilience of the nylon liner ensures that the bearing operates silently with a predetermined clearance: nylon wears at an extremely slow rate when used as a bearing for a steel journal. All these features reduce noise to a minimum. Generally, the nylon bushes are lubricated on assembly, and thereafter require no more attention. In fact, it is not necessary to use any lubrication at all with nylon, although the performance may be improved slightly by initial lubrication. The application of nylon bushes to vehicle chassis is still in the early stages of development, but it is expected that these components will prove to have a life equivalent to, for example, that generally obtained with metal bushes used in conjunction with automatic lubrication systems. In other words, by employing nylon bushes, it should be possible to eliminate complicated lubrication systems and obviate the need for attention between major overhaul operations.

Fig. 3. Curves showing the radial rate plotted against the wall thickness of the rubber of flexible bearings. The diagram is divided by the two dotted lines into three areas indicating the average tolerances



The question arises as to where nylon should be employed in preference to rubber. First, the space requirements of nylon bushes are more modest than those of rubber bushes. Secondly, nylon can be employed where rubber would be too flexible, for example, in certain steering and suspension joints. Thirdly, it can be employed where rubber bushes would be unacceptable because their torsional rate or radial and axial resilience would adversely affect the operation of the components to which they are applied. Finally, if the angle of oscillation of the journal is large, it might not be possible to accommodate it by means of a rubber bush of acceptable diameter.

Clayflex bearings

Clayflex bearings are among the products manufactured by Howard Clayton-Wright Ltd., of Wellesbourne, Warwickshire. They differ from most of the other rubber bearings available in that both chemical and pressure bonding are employed in their manufacture. When pressure bonding

Fig. 1. A typical example of a bush in which the rubber is pressure bonded to both inner and outer sleeves. The dotted lines indicate the size and shape of the rubber section before it is compressed between the sleeves

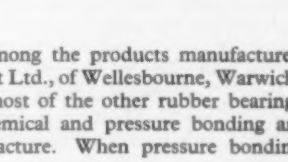
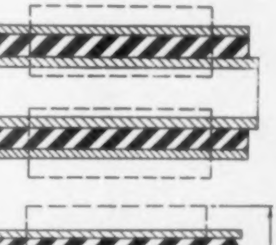


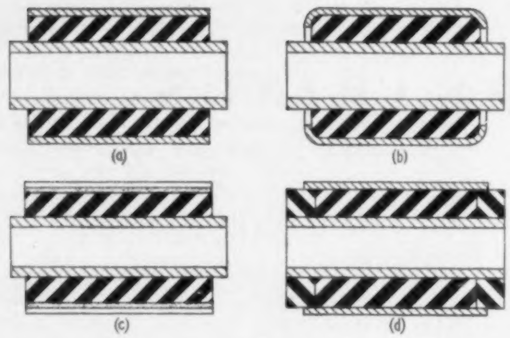
Fig. 2. Section through a Clayflex bearing, the rubber of which is bonded to the inner sleeve over the length L. The diameter D is that of the rubber before it is forced into the outer sleeve



alone is employed, the relative proportions of the rubber bush and its steel inner and outer sleeves are approximately as shown in Fig. 1. The initial dimensions of the rubber bush are indicated by the dotted lines, and precompression is effected by forcing the rubber through a tapered die into the space between the inner and outer sleeves.

The limiting factor of a wholly pressure bonded construction is the degree of frictional grip obtainable, between the rubber and steel sleeves, as a result of the radial forces exerted by the rubber in compression. When slip occurs, it is invariably between the rubber and the inner sleeve, since the contact area between the rubber and the outer sleeve is much larger. This restricts the potential capacity of bearings of the pressure bonded type.

In the Clayflex bearings, the inner sleeve is chemically bonded to the rubber during the vulcanizing process. Subsequently, this subassembly is forced into the outer sleeve, a tapered die being used to compress the rubber, so that the finished component is as shown in Fig. 2. In this illustration, the initial shape of the rubber is indicated by the dotted lines. By chemically bonding the rubber to the inner



a *the basic design, termed the B type;* **b** *the BP type, which is identical in all respects with the B type, except in that the ends of the outer tube are peened;* **c** *the BS type, which has a two-piece outer sleeve, and in which the rubber is bonded to both the inner and the outer sleeves;* **d** *the BE type, which has a rubber extension bonded to each end of the inner sleeve, the space between the three pieces of rubber being closed on assembly by the displacement of the centre portion under compression*

Fig. 4. The four standard designs of the well-known Clayflex bearings

sleeve and pressure bonding it to the outer sleeve, the manufacturers have increased the capacity of the bearing well beyond the point that was previously regarded as the operating limit for pressure bonded rubber bearings.

The advantages of this type of bearing construction are as follows. The operating loads are not borne entirely by the chemical bond. Should the bond fail for any reason, for instance, as a result of gross overload, the bearing will not necessarily cease to function, since the pressure bond remains operative. The characteristics of the bearing can be accurately predicted at the design stage and can therefore be easily varied to suit requirements. Bearings can be designed to cater for a wide range of torsional applications, including high frequency reversals. Under conditions of high frequency reversal, the chemical bond obviates the risk of the inertia forces overcoming the frictional grip of the rubber on the inner member. Excessive torsional deflection, due to momentary overload, does not destroy the bearing, since slip will take place between the rubber and the outer sleeve.

The radial load capacity of these bearings is high because the chemical bond restricts the flow of the rubber away from the pressure area. It can be further increased by peening radially inwards the ends of the outer sleeves. Peened outer sleeves are recommended for applications where the bearings are subject to axial load. The peening prevents relative axial movements between the rubber and the outer sleeve, while the bond prevents axial movement of the rubber relative to the inner sleeve. This type of bearing is particularly suitable for application where the load is high or is of a sustained or vibratory nature.

Conical deflection is automatically catered for in most cases, since it is usually of a secondary nature relative to the main loading. However, should the conical loading be considerable, and particularly if it is required to be a main function of the bearing, it should be considered carefully in relation to the maximum permissible radial deflection of the component.

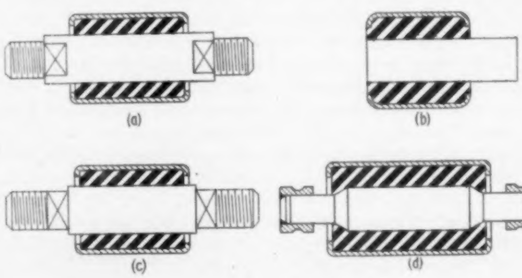
Practical considerations and designs

Curves showing the radial rate plotted against thickness of rubber for various outside diameters of the inner sleeve are illustrated in Fig. 3. These curves are divided by two dotted lines into three areas, the percentage figures indicate the average tolerances in respect of rate for the nominal dimensions in the area. From this diagram, it can be seen that, for example, if a bearing is required to have a strictly controlled rate, it should have a relatively thick rubber sleeve. This is because, as the thickness of rubber increases, variations in thickness have a decreasing effect on the rate of the flexible unit.

In general, the axial and torsional rates have an average tolerance of ± 10 per cent as compared with the figure of ± 15 per cent for the radial rate. Since these two rates also conform to a similar law to that of the radial rate, it follows that the smaller the thickness of rubber, the greater is the tolerance required. When bearings of the peened outer sleeve type have to be used, it must be borne in mind that peening cannot be effected if the rubber sleeve thickness is too small. Also, if the length of the outer tube is less than its diameter, then the length of the parallel portion between the peened ends generally is too small to be held effectively in the housing.

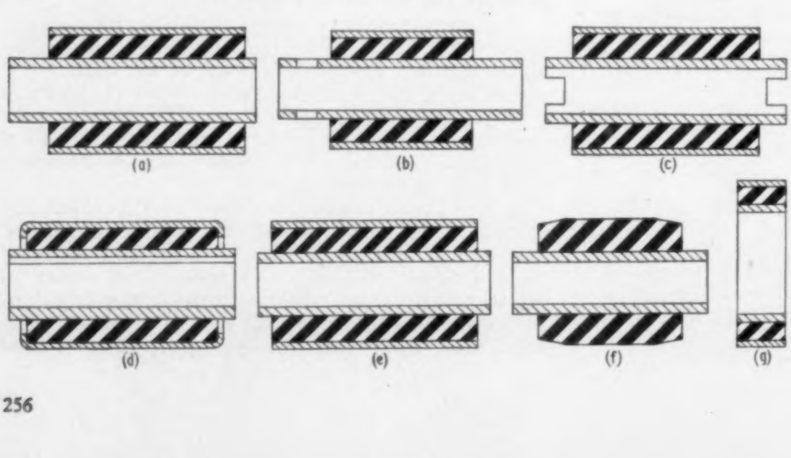
A number of special Clayflex bearings are manufactured, as well as four standard types. The basis of the four standard designs is the B type, Fig. 4a, which has plain inner and outer sleeves, with the rubber precompressed between them and bonded to the inner sleeve. A variant is the BP type, which is identical in all respects to the B type, except that the ends of its outer tube are peened. It is recommended for all applications where axial loads predominate. The peening of the outer member has the effect of increasing both the radial and the axial rates by approximately 10 per cent.

Thirdly, there is the BS type, in which the rubber is bonded to both the inner and outer members. To obtain



a *bearing with a central stud, on the ends of which are milled flats and threads;* **b** *asymmetrical stud type bearing;* **c** *ends of stud milled square and threaded;* **d** *grooved collars, for pinch bolts, on the ends of a stud of double-conical form*

Fig. 6. Above: Some of the stud type bearings made by Howard Clayton-Wright Ltd.



a *unequally extended ends of inner sleeve;* **b** *inner sleeve with location holes;* **c** *inner sleeve with location slots;* **d** *inner sleeve with keyway;* **e** *inner sleeve of oil-retaining material;* **f** *type SAB, which has no outer sleeve;* **g** *the very short type BB*

Fig. 5. Left: A selection of the special types of Clayflex flexible bearings

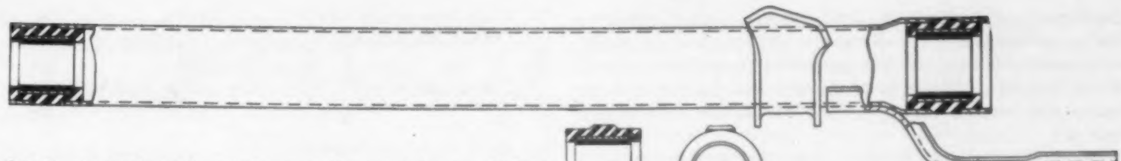
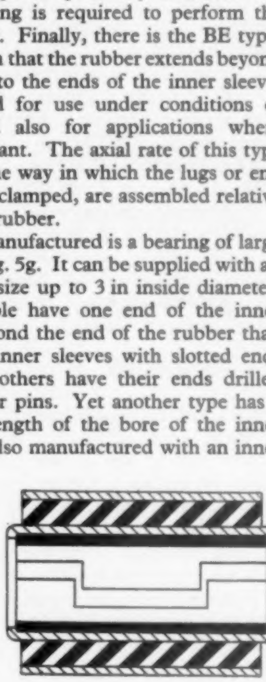


Fig. 9. A steering column tube assembly containing two Nylastic bearings, and, shown separately below it, an alternative bearing design with a steel stop-flange moulded in one end of the rubber housing

the precompression, the outer housing is split longitudinally in two, and the halves are compressed together on assembly into the housing. This type is primarily intended for applications where the bearing is required to perform the function of a torsional spring. Finally, there is the BE type. This differs from the B type in that the rubber extends beyond the ends of the outer sleeve to the ends of the inner sleeve. It is specially recommended for use under conditions of intermittent axial load and also for applications where electrical insulation is important. The axial rate of this type of bearing is influenced by the way in which the lugs or end washers, between which it is clamped, are assembled relative to the extended ends of the rubber.

Among the special types manufactured is a bearing of large diameter and short length, Fig. 5g. It can be supplied with an inner sleeve of any nominal size up to 3 in inside diameter. Other types of bush available have one end of the inner sleeve projecting further beyond the end of the rubber than the other end. Some have inner sleeves with slotted ends for positive location, while others have their ends drilled diametrically to receive cotter pins. Yet another type has a keyway machined the full length of the bore of the inner sleeve. Clayflex bushes are also manufactured with an inner

Fig. 7. On the right is a Clayflex flexible bearing with a Nylastic sleeve located in the inner metal sleeve; and, below, is a bearing with a Nylastic sleeve in a steel outer sleeve, the two being retained in a resilient rubber housing



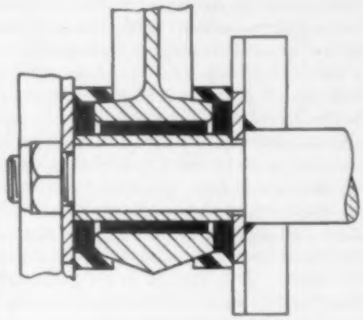


Fig. 10. Right: This bearing assembly is employed for the lower inner pivots of the Triumph TR3 front suspension. The Nylastic bushes and thrust washers are shown black, while the rubber sealing rings are hatched with alternate black and white bands

sleeve of oil-retaining material. For applications where economy in regard to both cost and space in the installation is a prime requirement, there is a Clayflex bush without an outer sleeve. This has to be pressed through a tapered die into its housing.

The Clayflex stud-type bearings are another variant in common use, Fig. 6. With these, the rubber is bonded on to a pin, or stud, instead of on to an inner sleeve. These all have peened outer sleeves, but differ from each other in respect of the shape of the pin or stud employed. Some have parallel shanks, with their ends milled square and threaded, while others have flats milled on their ends and are threaded. Bearings of this type can also be supplied with the pin chamfered just inside each end of the rubber sleeve, Fig. 6d. These chamfers, together with the lips formed by peening the ends of the outer sleeve, increase the axial load-carrying capacity. Collars can be welded on the ends of the pin and are grooved circumferentially to clear a pinch bolt in its split housing. Bearings in which the studs project further from one end than from the other are also made.

Nylastic bearings

Nylastic bearings are designed for carrying oscillating or rotating journals. Basically, they are simple in construction, comprising a steel outer tube, into which is fitted a Nylastic bearing liner, which is a type of nylon. The advantages of nylon for bearing material are well known; they are mainly its resilience and low coefficient of friction, even in the unlubricated state. For some applications, the nylon liners are manufactured from strip, and for others they are individually moulded. Each liner comprises a strip of material with a tongue projecting from one end and a slot at the other. When the strip is rolled round to fit inside the sleeve, the tongue registers in the slot to locate the ends relative to one another. A fairly large clearance is left between the abutting

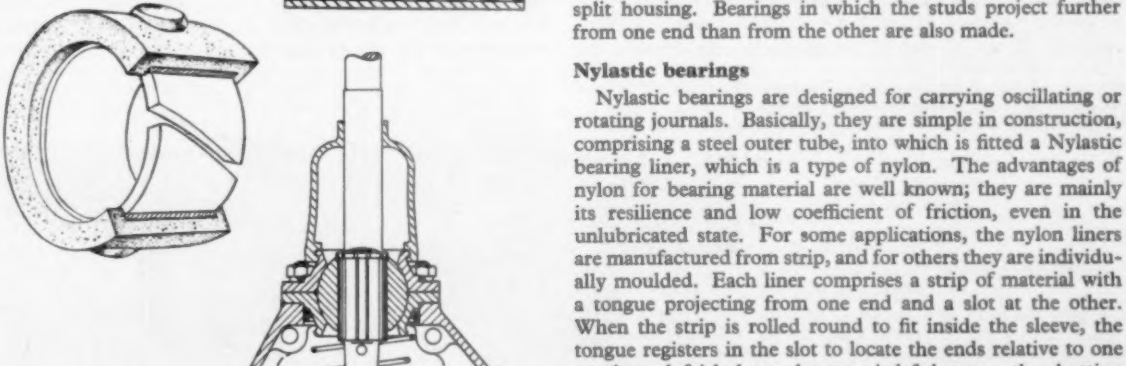


Fig. 8. Two Nylastic bearings are employed in this gear shift control mechanism. The Nylastic material is shown black in this illustration

shoulders, and between the ends of the tongue and slot. This clearance allows for changes in the length of the strip, which are liable to occur with variations in moisture content of the Nylastic. Variations of thickness of the strip are, of course, too small to affect in any way the operation of the bearing.

In some applications, the ends of the strip are finished at an angle instead of tongued and slotted, Fig. 7, but this arrangement is not recommended if the loading is heavy, since there is a danger that, because of the angle of the abutting faces, the Nylastic lining might be moved axially out of its housing by the rotary motion. Positive axial location can be provided for by the employment of Nylastic thrust washers.

These bearings are generally lubricated on assembly, with an oil such as EP740. Thereafter, they require no further attention. They have been found to operate perfectly satisfactorily at PV factors of 7,500, where P is the bearing pressure, in lb/in^2, on the projected area and V is the relative velocity, in ft/min, between the bearing surfaces. If the bearing is required to be located positively in the outer sleeve, small circular projections, or pips, are moulded on the outer surface of the liner, and these register in holes punched in the sleeve. This feature is shown clearly in the illustration showing the cross section of a gear shift control, Fig. 8. It is generally considered advisable to use bearings the length of which is no longer than the diameter. This is because Nylastic is a poor conductor of heat, and the temperature might become unduly high in bearings that are too long.

The thickness of the Nylastic material can be anything from about 0·020 in to approximately 0·060 in. If it is too thick, the material is liable to spread out of the housing. The smaller dimension quoted is adequate to allow for foreign bodies, which might enter the bearing, to embed themselves in the Nylastic, where they will not damage the shaft.

An interesting application for Nylastic bearings is the support of a steering column in its tube. As can be seen from Fig. 9, the Nylastic bearing assembly is housed in a rubber sleeve, both ends of which are lipped internally. The dimensions of these lips are such that they retain both the steel sleeve and the Nylastic liner. The outer periphery of the rubber housing has two circular section projections, or pips, moulded on it, diametrically opposite one another; these register in holes in the tube in which the whole assembly is housed. A steel washer can be moulded into the lip at one end of the rubber, if necessary, to ensure that the

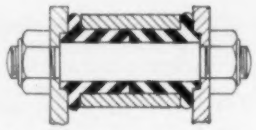


Fig. 16. Above: A modified Harrisflex bearing for applications where the axial loading is a major consideration

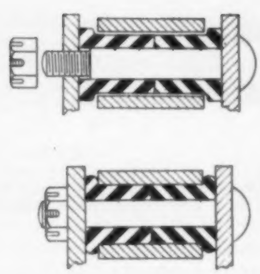


Fig. 14. Right: Two illustrations showing the basic principles of the Harrisflex bearing assembly

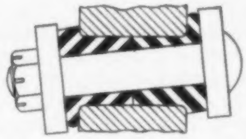


Fig. 15. Above: The flanges of the Harrisflex bearing help to react torsion about axes normal to that of the pin

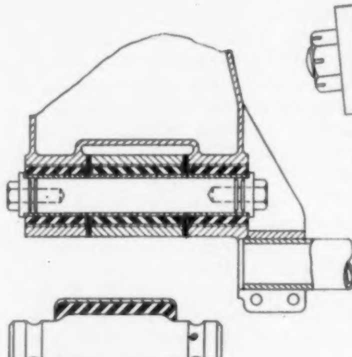
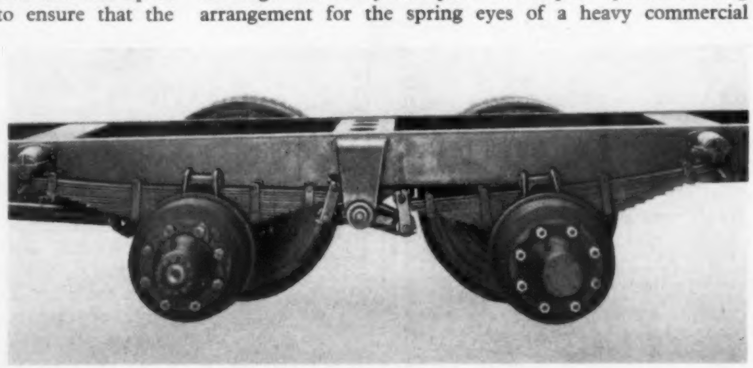


Fig. 13. Left: The Clayflex bearing and Nylastic thrust washer assemblies used in the balance beam assembly and shackles of the trailing axle suspension shown in Fig. 11

Nylastic bearing cannot be pushed out of the rubber when the assembly is pressed into the tube. The principal advantages of this type of bearing are its self-aligning characteristics, ability to absorb a certain amount of vibration, long life, freedom from rattles, and the fact that it does not require lubrication.

Three examples of the application of nylon to suspension bearings are shown in Figs. 10, 11, 12 and 13. In Fig. 10 is illustrated one of the lower, inner bearings of the transverse wishbone type front suspension on the Triumph TR3. Not only is the bearing bush of nylon, but also nylon thrust washers are fitted at each end, between the bearing boss and the lugs that carry the pin. A simple Nylastic bearing arrangement for the spring eyes of a heavy commercial

Fig. 11. Right: A trailing axle conversion for a Commer 7 ton vehicle, in which Clayflex bushes and Nylastic thrust washers are used



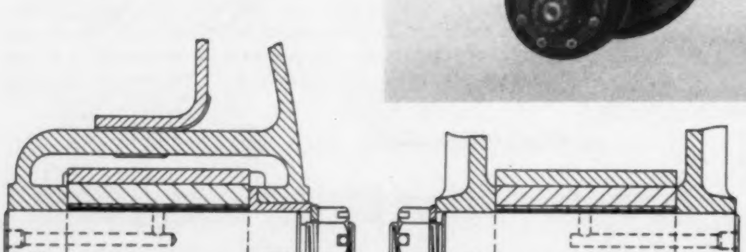


Fig. 12. Left: A simple Nylastic bearing arrangement for the eyes of semi-elliptic springs, as used in heavy commercial vehicles

vehicle is illustrated in Fig. 12. Figs. 11 and 13 show respectively a trailing axle conversion for a Commer 7 ton vehicle and details of the Clayflex bearings used in the balance beam assembly and shackles. Nylastic thrust washers are employed in the balance beam bracket.

Harrisflex bearings

Howard Clayton-Wright were the first to introduce Harrisflex bearings in this country. These bearings, as is well known, are of rubber and they are available in a wide variety of forms. The basic principle in all applications is the same, and is as shown in Fig. 14.

Harrisflex bearings of the spool type, that is, with the rubber supplied as one unit, are available, but they are more commonly made up of two moulded bushes. These two parts fit easily into the housing and over the pin. The free length of the bearing and the length of the shouldered shank of the bolt are carefully controlled to give the appropriate amount of preload when the end plates are tightened against the rubber. Close tolerances are not necessary in the manufacture of the metal components: machining is usually done to a tolerance of 0·005 in.

An important advantage of the Harrisflex bearing is that, should the assembly be subjected to torsional wind-up for any length of time, the bearing will begin to creep until it is returned to its neutral condition, as assembled. Thus, it is not necessary, on assembly, to fit the bearing in any specified angular relationship to its shaft or housing, since it will automatically adjust itself to the mean position of the linkage.

The efficient functioning of the bearing is, to a large extent, dependent on the performance of the shoulder, that is, the portion that flows between the end faces of the housing and the end plate when the precompression is applied. This shoulder should not be confused with the moulded flange on some types of bearing. The formation of the shoulder under pressure ensures that the natural elasticity of the rubber is used to prevent movement relative to the metal components during normal operation. It is not always realized that the shoulder helps to react twist about all three planes, as well as axial force. The way in which twist about axes other than that of the pin is reacted can be seen from Fig. 15 on the opposite page.

For applications where the axial forces are extremely high, the modification shown in Fig. 16 has proved to be most effective. This modification is simply the counterboring of the holes in the end plates to receive the projecting ends of the rubber bush. It is a recently patented feature, which imposes a limitation on the flow of rubber into the shoulder and thus has the effect of increasing both the pressure inside the housing and the tension on the shoulder. As a result, a lower rate of deflection, both axial and radial, is obtained under load. An added advantage is that wider shoulders can be employed without danger of instability, and this increases the maximum practicable angle of oscillation of the pin relative to its housing.

The advantages of the Harrisflex bush can be summarized as follows. Assembly can be effected easily and rapidly, as also can replacement, should this be necessary. Close tolerances on the metal components are not necessary. Provided the bush is correctly applied, long and trouble-free service life is obtained. After the initial fitment, no further adjustment is required. Like most flexible bearings, this type does not require lubrication. It is silent in operation and is effective as a vibration insulator. Special designs to meet even the most exacting specifications can be produced.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Plastics Progress 1957

Edited by Philip Morgan, M.A.

London: ILIFFE AND SONS LTD., Dorset House, Stamford Street, S.E.1. 1958. 9¾ × 6. 394 pp. Price 50s.

This collection of papers presents some of the most recent international advances in plastics technology, particularly in the rapidly expanding fields of polythenes, vinyls, polystyrenes, reinforced plastics and fluorine polymers. Also dealt with are current developments in the theory and practice of extrusion and injection moulding. The authors are leading authorities from the United Kingdom, the United States and Germany, and much of the information contained in their papers is based on original work not previously published.

All the contributions were the subjects of papers at the 1957 International Plastics Convention, held at Olympia, London, where they were presented in an abbreviated form. This volume permits the authors to develop their theses to lengths not possible in the limited time available at the Convention itself, and therefore provides a valuable and, in many respects, a unique source of information on plastics technology.

The headings of the chapters are as follows: Polyolefins; Polythene; Polyvinyl Chloride; Extrusion; Injection moulding; Material developments; Glass reinforced plastics; a subject index is also included.

Casting in Steel

London: SIR ISAAC PITMAN AND SONS LTD., Pitman House, Kingsway, W.C.2. 1958. 9 × 6½. 112 pp. Price 18s.

This, the first edition of the book, is published on behalf of The British Steel Founders' Association. One of its objects is to show how steel castings can, with advantage, be applied over wider fields of engineering construction than at present. The work, therefore, is aimed at informing engineers and designers as to the characteristics and functional attributes of cast steel components likely to be of use in a wide range of applications.

The work is based on the BSFA bulletins. Some of these bulletins, particularly among the earlier ones, have been revised in the light of current knowledge and, together with more recent ones, some of which are now out of print, have been collected together to form this volume. The work has been carried out under the supervision of a group of expert steel founders. This book, therefore, supplies information about the process of casting steel in component form without which it is not possible to make the best and most economical use of this production technique for the manufacture of steel parts. The work may also serve as an introduction to the steel founding process for those beginning their studies in this field.

Trouble-Free Hydraulics

By Ian McNeil, M.A., A.I.Mech.E.

London: THAMES AND HUDSON LTD., 30 Bloomsbury Street, W.C.1. 1958. 8¾×5½. 124 pp. Price 18s.

This is a practical book for engineers who have not time to study the theoretical aspects of hydraulics: that is, for those for whom a knowledge of how to keep hydraulic machinery running efficiently, and what to do if it goes wrong, is essential to their work. Among the subjects dealt with are: the laying down and carrying out of preventive maintenance schedules; starting up new machines; anticipated life of hydraulic oil, and when to change the oil; the danger point in solid contamination of oil; care of oil seals and replacement; and diagnosis and cure of faults.

A great deal of the information has been given in tabular form, and this is cross indexed for rapid reference. Also, a comprehensive set of fault-finding charts is included, to enable machine troubles to be overcome quickly, even by those whose knowledge of the machines and their fundamental principles is not very great. At the end of the work, there is a useful appendix of safe pressures in tubes and conversion factors.

THE SYKOMATIC GEAR GENERATOR

A Magazine-Loading, Turret-Indexing, Automatic-Chucking Machine for Large Batch or Continuous Production

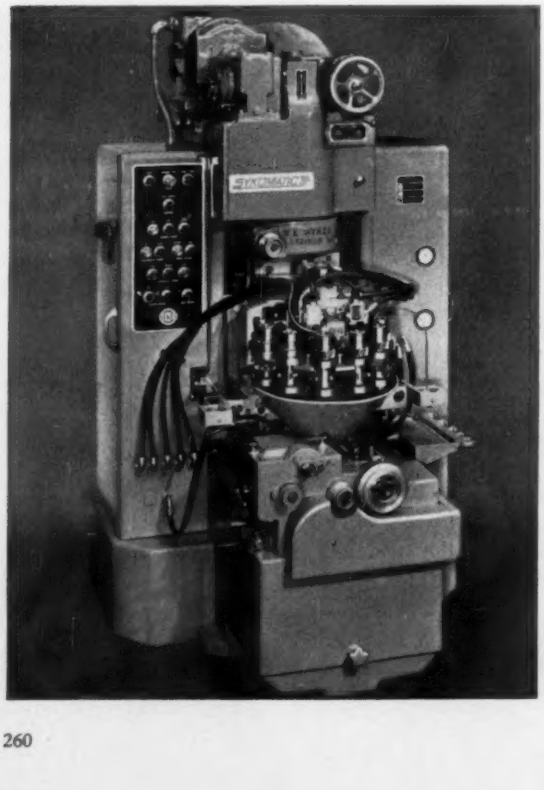
A NEW machine developed by W. E. Sykes Ltd., Manor Works, Staines, Middlesex, offers substantial practical advantages to producers of large quantities of similar gears. Its outstanding characteristic is the ability to bring virtual automation to the gear-cutting shop without requiring the heavy capital outlay of an automatic transfer mechanism or occupying an inordinate amount of shop space. Each machine is completely self-contained but, of course, they can be readily linked by track or transfer equipment if desired.

The machine is equipped with angular cutter relief, and an automatic loading device fitted with a rotary turret magazine to hold a number of components. The model shown has a magazine to hold twelve automobile layshaft gears.

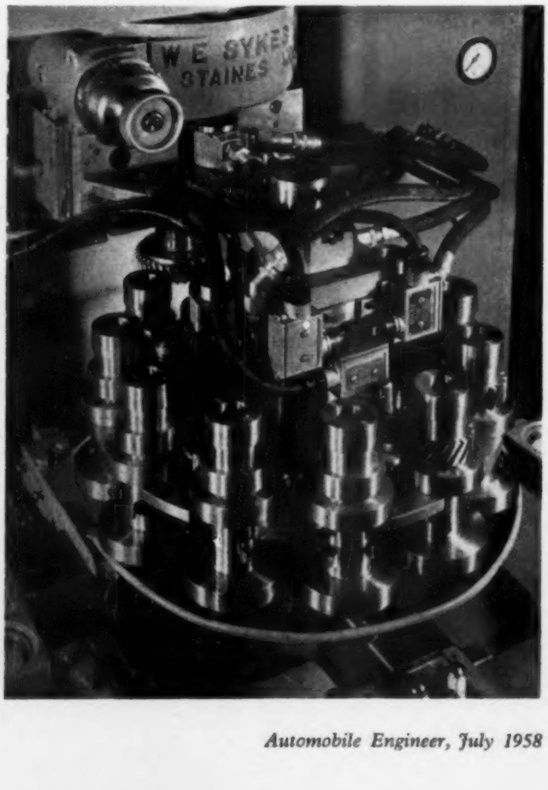
The turret, which is loaded with a full complement of blanks, permits fully automatic and uninterrupted operation of the machine for nearly 40 minutes, the cycle time for each gear being approximately 3·2 minutes. Loading is particularly simple, as the blanks are quickly pressed into the spring clips. Apart from the obvious facility of being able to employ unskilled operators, the automatic loading arrangement enables one person to maintain full output from a number of machines.

The main fixture base is bolted to the saddle, and a heavy vertical column from this supports the indexing magazine and a top plate. The latter, which carries the tailstock and indexing piston, is fixed to the top of the column, both elements being supported by a slide bolted to the front of the machine. A hydraulic piston and a dead stop are provided to lock the fixture at the correct centre distance position.

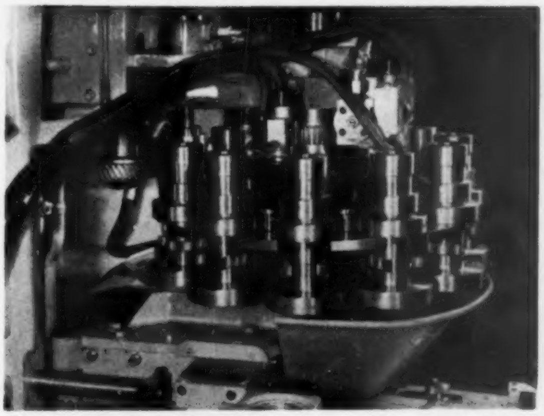
The magazine is free to rotate and slide on the column. A spur gear is fixed to the upper part of the magazine, and rack teeth are cut in the indexing piston rod. The chuck is of the expanding mandrel type and is bolted to the worktable. With the magazine fully loaded, the index piston forward, magazine up and chuck released, the cycle start button is pressed. The tailstock moves down to enter the bore of the component and continues its downward movement, taking the magazine down with it and pushing the work on to the chuck. When the gear is hard against the locating face of the chuck, the latter is operated by a sequence valve to clamp the work. The index piston moves to the rear, cutter and coolant motors start, the component is fed towards the cutter and the normal gear shaping operation commences. During infeed of the saddle, oil at low pressure is applied to the clamp to give additional support to the fixture. When the gear is at the correct cutting depth, oil at high pressure is applied to the clamp and the fixture is locked for the final cut. After the gear has been cut the clamp is released, the saddle retracts and the cutter motor stops. The chuck is



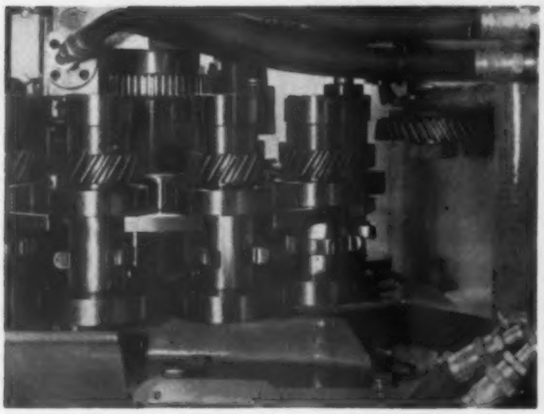
The Sykomatic magazine-loading gear generating machine is completely self-contained



General view of indexing magazine set up for transmission layshaft gears



The work is easily loaded into the magazine and retained by spring clips



Detail of the rotary magazine, showing the indexing gear and the cutter

released and the tailstock and turret move up to lift the component clear of the chuck. The tailstock continues upwards until it is clear of the work, when the index piston moves forward. The rack teeth engage the magazine spur gear and index the next component to the loading position, when the cycle restarts.

A "continuous/single" switch is provided. At "continuous" the cycle repeats, and at "single" one gear only is cut. If the switch is turned to "single" during cutting, the machine will stop at the end of that cycle.

Independent buttons are provided for all motions to facilitate setting up, and these are interlocked through a "cycle/manual" switch to ensure that they are inoperative when the machine is on automatic cycling. Similarly the cycle start button cannot be used when setting up. A red light indicates that the hydraulic pump motor is energized.

Two changeover switches are fitted, one to select high or low cutter speeds and the other to isolate the coolant pump motor when setting up. A mushroom-headed emergency-stop button cuts out the saddle infeed, the cutter motor and the coolant pump motor, but the hydraulic pump motor continues to operate. The control cabinet isolating switch is located adjacent to the control panel and the position of the large operating handle indicates whether the power supply is on or off.

Electric starters and relays are of the "plug in" type and it is necessary to remove only two screws to detach a unit without disturbing the wiring. All the electrical control gear is contained in the cabinet and a safety interlock switch ensures that the supply is cut off before the oil- and dust-proof door can be opened.

The hydraulic control valves are gasket mounted, and are removable by unscrewing four bolts; no piping is affected. The oil reservoir can be easily detached, and a large cover plate provides complete access for cleaning the reservoir and filter. A vane-type pump supplies oil at pressure, and the system is safeguarded by a relief valve. The sequencing of the various pistons is effected by limit switches controlling solenoid-actuated valves, and the lowered pressure for the clamping piston is provided by a pressure-reducing valve.

A typical component, cut on the Sykomatic, is illustrated and described here. Twelve layshaft gears are carried in the magazine, and the chucking is by an expanding mandrel. However, the form of gears intended for cutting on this machine may vary widely from that shown.

Gear data

20 teeth; 30 deg R.H. helix angle; 11·547 N.D.P.; 17½ deg N.P.A.

Cutter data

42 teeth; 30 deg L.H. helix angle; 4·2 in P.C.D.

Protuberance cutter also modified to provide 0·005 to 0·010 in chamfer at tips of component teeth.

Cycle data

2 Cuts; 413 strokes per minute; 0·875 in stroke; 0·014 in rotary feed per stroke; 3·2 minutes cycle time.

The general specification of capacities and equipment of the standard machine is as follows:

Maximum face width	2¼ in (57 mm)
Coarsest pitch	6 D.P. (4·25 Module)
Finest pitch	64 D.P. (0·4 Module)
Cutting stroke per minute	149, 186, 224, 280, 330, 413, 498, 622 (an alternative slower range is available)
Rotary feed per stroke	0·0005 to 0·030 in (0·013 to 0·762 mm)
Cutter motor	2 h.p. 2-speed (1,440/930 rev/min)
Hydraulic pump motor	1 h.p. (960 rev/min)
Coolant pump motor	½ h.p. (1,420 rev/min)
Floor space	48×58 in (1,220×1,470 mm)
Height	81 in (2,060 mm)

The maximum diameter of gear that can be cut is dependent upon the nature of the work.

AIRLOCK TUBE

An inner tube, termed the Airlock tube, has been introduced recently by Pirelli Ltd. It is made of Butyl, which, unlike rubber, does not have any affinity for oxygen, so that loss of pressure due to diffusion caused by the oxidation of the tube material is obviated.

The manufacturers have developed new production techniques to exploit the potentialities of the material. As a result, the new tube is thicker and of more uniform dimensions than those manufactured hitherto. A feature of the production process is the special precautions taken to avoid thin patches developing in the tube during the handling and shaping operations before curing. High quality is ensured by comprehensive inspection at every stage in the production.

The joint in the Airlock tube is not overlapped, and as a result, a clean, extremely strong and almost invisible joint is obtained. A new method of attaching the valve has also been evolved; this gives an extremely good bond and a well finished appearance.

Car Aerodynamics

Part II: The Second Series of Tests Carried Out at Ardmore, Analysis of the Results, and the Outlook for the Future

G. E. Lind Walker, M.A., A.F.R.Ae.S., Assoc.I.Mech.E., A.M.N.Z.I.E.

AFTER the tests described in the first part of this article were completed, further work was done with the object of analysing the results obtained from car 1 and of evolving a shape with satisfactory aerodynamic characteristics. For the later tests of this series, yawing moment and side force measurements were made with the model supported by an overhead claw. This enabled far more accurate measurements to be made. The tail sting mounting and its support cradle were retained for lift and pitching moment measurements. From the outset, the work was done with solid shapes, as it was thought that these would provide unambiguous results. The first of these models, shape 1a, is shown in Fig. 16. It is simply the car 1, with the wheels removed and all openings sealed smoothly with Plasticene. All the three directional components for this shape are shown in Fig. 17.

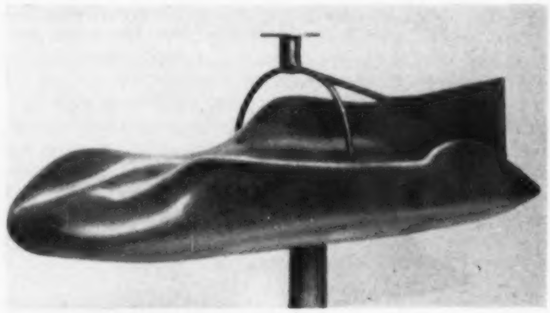
The resistance $C_w = 0{\cdot}045$, is very low, but the directional characteristics are clearly still bad. This test, which was made with the tail sting model support, shows the wide scattering of the points observed for the yawing moment. Side force observations indicate a slight lack of symmetry of the model.

The next step was to increase considerably the fin area, the tail of the body being elongated to suit. As a result, shape 1b, which is illustrated in Fig. 18, with the claw of the overhead mounting attached, was produced. The claw has a considerable effect, particularly on the resistance of the model, for which correction has been made; the yawing moment of the claw and its support shaft is very small, even compared with that of the model. In Fig. 19, the six component coefficients for this shape are shown.

Shape 1b was also employed for some other tests, as follows. First, it was mounted in the free flight condition, that is, without the road beneath it, and a test was made over a range of angles of incidence, α, as the nose was raised and lowered. The results, given in Fig. 20, show a discontinuity of the curves between $\alpha = -6$ deg and $\alpha = -9$ deg. This indicates a change of the flow pattern, presumably in the form of a stall or breakaway from the surface, at larger values of α, where the value of C_d is greater.

Fig. 16. Above: The first of the models, shape 1a, used in the second series of tests is simply the original car 1 with the wheels removed and all openings completely sealed and faired smoothly with plasticene

Fig. 18. Below: Shape 1b, with the claw of the overhead mounting attached; the experimental results have to be corrected for the effect of the claw. On this model, the fin area is larger than on shape 1a



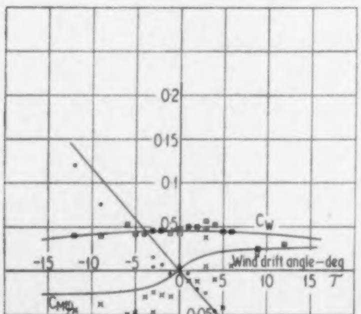


Fig. 17. Resistance and side force coefficients and yawing moment coefficient, based on the wheelbase area and referred to the midpoint of the wheelbase, at a height of 0·198l above the ground, in the manner defined in Fig. 13

The shape was then set in its normal attitude, with $\tau = 0$ and $\alpha = 0$, and the road was gradually raised beneath it. The results are shown in Fig. 21. It is of interest to note that, at a value of g/l of about 0·02, the lift force began to increase fairly rapidly, and at the same time the model started to flutter.

Reduction of the tunnel speed for the final test readings prevented further flutter. Therefore, it is open to speculation as to whether such a condition of aerodynamically energized oscillation could develop at high speed in a real car, or whether the suspension damping would prove adequate to deal with the energy input. At the normal ground clearance level, the coefficients of this shape were hardly different from the free flight values, which are indicated by arrows at the edge of the diagram.

An investigation was also carried out on the effect of undersurface roughness. Fig. 22 shows the arrangement of square bars fitted across the bottom of the shape, and the effects that they produced. The change produced in respect of lift

and pitch was generally large, while the change of drag remained small for each arrangement. In this test, flutter also was observed, but only with some of the arrangements.

Shape 1c was derived from shape 1b by thickening the side of the fin, so that the combined fin and cockpit canopy formed a normal aerofoil section. This shape was tested for only yaw and side force.

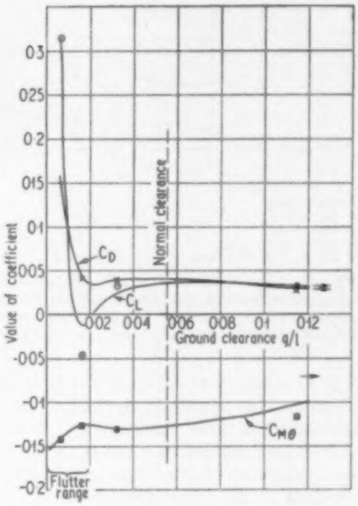
Confirmation of the poor directional characteristics of these shapes led to a careful study of the other published works. Sawatzki, whose investigation had been primarily concerned with directional stability, proved particularly helpful. No reference was found to marked non-linearity of the yawing moment curve at small drift angles; perhaps it is not usual. At larger drift angles, the curve is commonly non-linear and, in fact, unfinned bodies are normally stable at about $\tau = 90$ deg. Sawatzki points out that large, destabilizing moments are generally associated with long, slim, streamlined bodies, particularly with downwardly sloped tails. Cutting off squarely the tail of such a body greatly decreases the yawing moment for only a very moderate increase of resistance. Two other points are also indicated:

1. Location of a fin in a region of disturbed flow, for example, behind an open cockpit, greatly reduces its effectiveness.
2. Short-chord fins, with the lesser movement of their own centres of pressure, are more effective, at small drift angles, than large-chord fins. This leads to the consideration of numbers of small fins, which may be so located as to form slotted groups.

At this stage, the problem becomes clearly one of locating short-chord fins on the rear of the body in such a way that they are outside the wake from the open cockpit. On the other hand, the large lateral component of the streamlines around the tail must also be taken into consideration. If the airflow pattern is envisaged as that of the composite free stream body of the car and mirror image, it can be seen that fins radial to this free stream body would fulfil both the above requirements. The V-tail, of shape 1d, was then designed. It comprised two straight, symmetrical aerofoils of NACA 0015 section fitted to the rear of the body, inclined outwards at 20 deg, one each side of vertical. This tail was built up on the model, after the original central fin had been cut down until only a fairing behind the cockpit remained.

Resistance, side force and yawing moment coefficients of the four shapes are compared in Fig. 23. Too much significance should not be attached to the resistance figures, as the shapes were created with successively increasing areas of Plasticene with inferior surface finish; it is sufficient to note that there is no great change of resistance at any stage. It had

Fig. 21. Curves showing the effect of variation of ground clearance on the drag, lift and pitching moment coefficients of the car shape 1b



been thought that the tail of the original shape was swept up too rapidly away from the road, so the derived shapes were all lower at the tail, as can be seen by comparison of Fig. 11 with Fig. 25. The reduction of lift for all the later shapes indicates that the tail shape 1a was, in fact, so high that flow breakaway occurred. It is further deduced that, with equal surface finish, the later shapes should show some decrease in resistance. Shape 1d has a yawing moment curve of stable slope in the region of $\tau = 0$, which is the most important part.

At this stage, the basic shape was considered satisfactory, and a transformation was effected, step by step, back to the state of a normal open racing car. The first step was to refit the wheels, making car 1e. At this stage, the wheel arches were still carefully sealed with Plasticene. The next stage, car 1f, is shown in Fig. 24. This car has an effective cooling system with an air intake at the nose and discharge slots behind the rear wheels. The choice of exit arrangement was dictated by the construction of the original model, in which the cooling air passed through ducts, to be discharged beneath the rear wheel arches; it is an exit arrangement suitable for rear-mounted cooling installations, but otherwise is not particularly attractive. To make car 1g, the cockpit was opened; then the wheel arches were opened out to the original form, making car 1h, which is illustrated in Fig. 25.

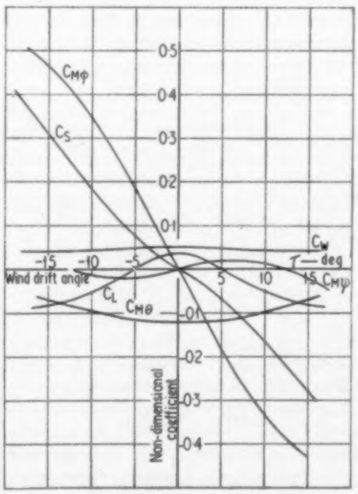
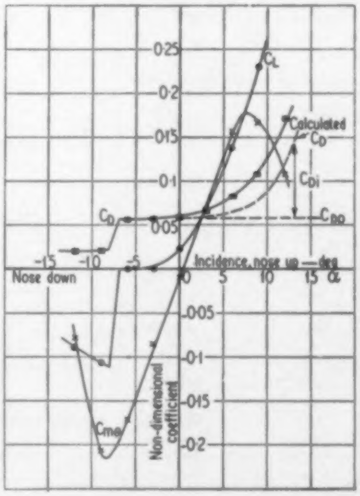
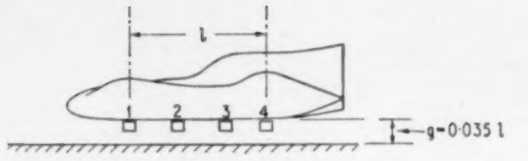


Fig. 19. The six component coefficients, for the shape 1b, based on the wheelbase area and referred to the mid-point of the wheelblase, at a height of 0·198l above the surface of the road

Fig. 20. Variation of incidence of shape 1b, mounted in the free flight condition. The discontinuity between α = −6 deg and α = −9 deg indicates a change in the air flow pattern





	C_D	C_L	C_{MO}
a clean model	0·074	0·033	—0·125
b bar 4 fitted	0·0747	0·211	—0·0974
c bar 1 fitted	0·0878	—0·0332	—0·0378
d bars 1 and 4 fitted	0·1121	0·0812	—0·0254
e bars 1, 2, 3 and 4 fitted	0·1030	0·0646	—0·0187

Fig. 22: This illustration shows the arrangement of square section bars, fitted beneath the model, for the investigation of the effects of under-surface roughness. The dimensions of the bars are 0·22l square section × 0·64t long, where l and t are as defined in Fig. 13, in Part I

The final state was car 1j, in which the front wheel ventilating passages also had been opened. The resistance figures for the model at these various stages are detailed in Table I.

The six components of the open car 1j are plotted in Fig 26. An unexpected feature was the drag reduction given by the opening of the brake cooling ducts. It might not be found in the real car, where suspension and steering gear form a much greater obstruction to airflow through the wheel arches. As might be expected, the resistance differs only slightly from that of the original model, car 1, but the other characteristics of this final development are satisfactory.

ANALYSIS OF THE ARDMORE TEST RESULTS

Resistance

There is a wide background of information on this aspect of car aerodynamics. Earl[13] gives figures for the air resistance of the pre-War German racing cars, which are collected in Table II, and presented in accordance with the convention used in this work. The exact test conditions on which these figures are founded have not been defined; nevertheless, from their values it is clear that the figure of $C_w = 0{\cdot}168$ obtained for car 1 is unduly high for an enclosed wheel design. This figure is, in fact, of the same order as the exposed-wheel Grand Prix cars of the 1939 era, and is much higher than that of the streamline cars of the same period.

Partial assessment of the detail constituents of resistance can be made by comparison with the figures given by Kraus for the Mercedes-Benz W-196: these figures are given in Table III, where it can be seen that, with a resistance coefficient of $C_w = 0{\cdot}1430$, the Mercedes-Benz model is considerably better than that tested at Ardmore. Even in the development state, when the wheel arches were completely sealed on the wheels, car 1g gave a value of $C_w = 0{\cdot}160$.

Comparison of the two bodies shows that the W-196 is considerably wider at the base, so that its wheels are much less exposed to frontal view. Further consideration of car 1e shows an increment $C_w = 0{\cdot}072$, due to the protruding portions of the wheels. This figure is disproportionate, since it is more than that of the complete body of the vehicle.

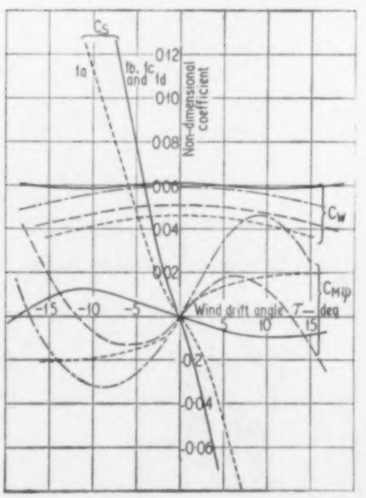


Fig. 23. Resistance, side force, and yawing moment coefficients plotted against the wind drift angle

TABLE II

Car	A_f ft²	C_d	l in	t in	S_w ft²	C_w
•Auto Union, open G.P. car	10·98	0·635	104	55	39·7	0·176
Auto Union, part faired record car	13·71	0·395	104	55	39·7	0·137
Auto Union, fully enclosed	16·0	0·280	104	55	39·7	0·113
Auto Union open G.P., body alone	6·35	0·057	—	—	39·7	0·0091
Mercedes-Benz, separate faired wheels, record car	14·0	0·21	107	58 51·5	40·8	0·072
Mercedes-Benz, open G.P. car, 1937	13·0	0·65	110	58	44·3	0·190

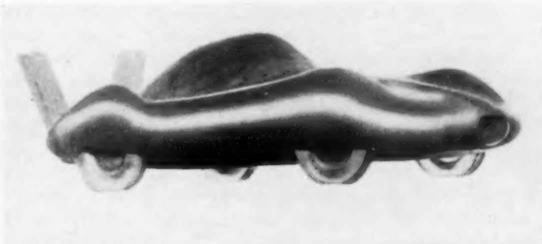


Fig. 24. Model car 1f, incorporating cooling system ducts, with an air intake at the front and discharge louvres behind the rear wheels

It is concluded that the body shape of really high-speed vehicles should be such as will provide maximum envelopment of the wheels. From Table II, it can be seen that the partially-faired Auto Union model, with an open cockpit,

TABLE I

Car	Resistance C_w	Increment added ΔC_w	Per cent total
1d	0·061		35·2
		wheels 0·072	41·6
1e	0·133		
		cooling 0·005	2·9
1f	0·138		
		open cockpit 0·022	12·7
1g	0·160		
		wheel arches 0·027	
1h	0·187		
		brake cooling —0·016	7·6
1j	0·172		100

has much the same resistance as the W-196, but the most noteworthy figure is that of $C_w = 0{\cdot}072$ for the Mercedes-Benz record car with separately faired wheels; this figure may be based on a test of a solid model without cooling system or internal flow passages, but even so, it can only be compared with the value of $C_w = 0{\cdot}061$ for shape 1d, which was even stripped of wheels.

Lift and induced drag

A moving car may be regarded as a lifting aerofoil, although of very low aspect ratio and subject to interference effects from the ground passing in close proximity below. The lift coefficients obtained in the tests varied, as between the different model stages. Car 1 had the greatest lift, which was coupled with a nose-up pitching moment; therefore, at the limit, the car would lift at the nose. The calculated speed at which the nose lifts is over 400 m.p.h., so it would appear that there is little danger associated with small positive values of the lift coefficient.

A consistent trend for the lift to decrease at larger values of the drift angle can be detected throughout the various development stages. This is unusual. Both Barth[14, 15] and Romani[9] produce a number of curves of C_L plotted against τ, not one of which has a negative slope in the region of $\tau < 30$ deg; in fact, many show a rapid rise. Possibly this common trend might be explained by the increase in effective aspect ratio, and thus in aerofoil efficiency, as the body assumes an attitude with its long axis across the wind.

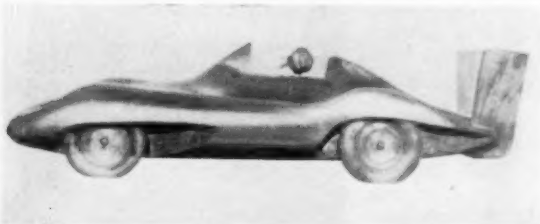
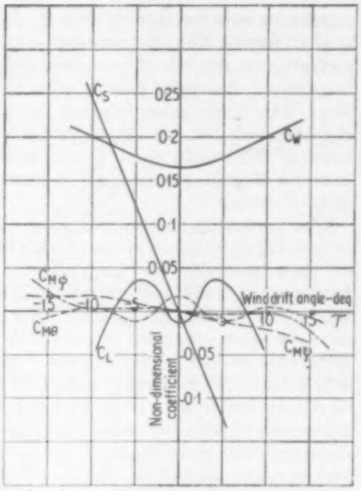


Fig. 25. Car 1g, which is a fairly advanced stage in the transformation from a solid model to a shape more representative of the racing car: the cockpit is open and the wheel arches properly formed round the wheels

Barth has investigated the normal generation of lift by a car body shape. The natural form approximates to that of a thick, heavily cambered aerofoil, with its flat, lower surface towards the road. Fig. 27, which has been reconstructed from Barth's work, shows how increased curvature of the lower surface of the shape reduces the effective aerofoil camber, and consequently the lift, at a constant angle of attack. In practice, this feature would be attained by sweeping up the ends of the body. In this context, examination of the basic form of the Ardmore model shows that its longitudinal centre line profile has only a small positive camber, as the ends are slightly elevated. In cross section, the profile is distinctly swept up at the sides; this gives, in effect, a considerable negative camber, which in this particular case accounts for the generation of negative lift, or down-thrust, which increases at large drift angles.

A car cannot be considered as an independent aerofoil, so far as designing for low lift is concerned, since the flow pattern is that of the composite shape, which includes the mirror-image beneath the road. These two parts of a composite aerofoil might have separate coefficients far greater than could be obtained by a single, complete aerofoil —a similar case is the slotted wing, where the slat, considered separately, often has a lift coefficient far higher than that obtained for the complete wing. The design target might be interpreted as the regulation of the pressure in the venturi passage that separates the two segments of the shape to about

Fig. 26. The six component coefficients of car 1j. An unexpected feature is the reduction of drag, owing to the opening of the brake cooling ducts



the same level as that around the outer surfaces. There will be no lift if, along the whole length of the model, $\Sigma p_{upper} = \Sigma p_{lower}$. Lift is always a direct danger at extremely high speeds, when the car might take off. Even at much lower speeds, particularly if the lift is coupled with a pitching moment, it may modify the tyre load distribution, with resultant change in the directional characteristics of the car, or perhaps reduced road adhesion.

Lift inevitably is accompanied by induced drag. Aerofoil theory provides a value for the induced drag, in the well known form:

$$D_i = \frac{L^2}{\pi \rho \, (2b)^2}$$

This is for conditions of optimum lift distribution. It can be conveniently rearranged as follows to give the induced drag coefficient:

$$C_{di} = \frac{C_L^2}{e \pi A}$$

and the total drag coefficient is:

$$C_d = C_{do} + C_{di}$$

where C_{do} is the residual drag, and has the value of C_d when $C_L = 0$. The term e is a correction factor for conditions of lift distribution other than ideal. Aspect ratio $A = (2b)^2/S_w$. In this particular case, the maximum y ordinate of the shape was taken as the semi-span b, which made $A = 0{\cdot}995$.

An allowance of $e = 0{\cdot}90$ has been made, and the induced drag of shape 1b has been calculated and plotted over a range of angles of incidence α, in Fig. 20. Though not actually

TABLE III

Item	C_w	Per cent
Model car, open cockpit and wheels fitted, as tested in wind tunnel	0·1161	80·9
Engine cooling system	0·0114	7·9
Suspension and steering gear protrusions	0·005	3·4
Driving mirrors and additional windscreen	0·005	3·4
Ventilation for cockpit and wheel arches	0·0036	2·4
Under-surface fittings and leakage	0·0029	2·0
Actual Grand Prix car	0·1440	100

coincident with the measured drag, the calculated line had a similar form. The discontinuities in the observed curves indicate that there is almost certainly a flow separation and consequent change of flow pattern within the range of this test. This could easily account for the difference between the observed and the calculated resistance curves. On the basis of this result, there is no reason to suspect that the induced drag of a car is not represented by conventional aerofoil theory.

The preceding analysis has been made for the condition of free flight, which of course differs from that of movement in close proximity to the ground. Interference of flow between the body and its mirror-image, which is another way of expressing the ground effect, does not yet appear to have been analysed theoretically in a form directly applicable to cars. Von Karman[16] has analysed the interaction between two lifting bodies: he shows that each tends to reduce the velocity over the other, thus producing losses in respect of both lift and drag. The effect varies with the distance apart, and there is a sudden reversal when the vortex centres come very close together. Toms[17] reproduces some tests of pairs of non-lifting struts. These show that the resistance increases as the gap between them is reduced, and this increase becomes very rapid as the gap is eliminated altogether. The motor car is in general a large body, producing relatively little lift, so that the latter case would seem the more applicable; however, Von Karman's work would suggest that the relationship between lift and induced drag remains unchanged in these conditions.

Flow visualization tests, as illustrated by Fig. 1, led to the investigation of the vortex strength behind the full-scale car. A free-running paddle wheel, shown in Fig. 28, was used to measure the circulation strength around its own axis. The measured vortex circulation strength was $\Gamma = 2{\cdot}05u$ ft/sec, where u is the car speed in feet per second. With the full-scale car, it was possible to evaluate the induced drag, which represents the energy carried away by the trailing vortex system. Von Karman has shown that the total energy in the trailing vortices can be expressed by mathematical equations as follows:

$$E = u\,D_i$$

and further, the induced drag is:

$$D_i = \frac{L^2}{2\rho u^2 \pi b^2}$$

while the lift is:

$$L = \rho u 2b\Gamma$$

A combination of these equations gives:

$$D_i = \frac{2\rho\Gamma^2}{\pi}$$

and:

$$C_{di} = \frac{D_i}{qS_w} = \frac{4\Gamma^2}{\pi u^2 S_w}$$

The experimental values obtained in this instance were:

$$C_{di} = \frac{4(2{\cdot}05u)^2}{\pi u^2\, 46{\cdot}9} = 0{\cdot}114$$

Since the wheelbase area was 46·9 ft², the corresponding lift coefficient was:

$$C_L = \frac{2b\rho u\Gamma}{\frac{1}{2}\rho u^2 S_w} = \frac{6u(2{\cdot}05u)}{\frac{1}{2}u^2\, 46{\cdot}9}$$

If the lift distribution factor is again taken as being $e = 0{\cdot}9$, the effective value of the lift coefficient is reduced to:

$$C_L = 0{\cdot}526\ \sqrt{0{\cdot}9} = 0{\cdot}500$$

This experimental work provided an actual value for C_{di}, although careful consideration indicated that all the probable errors lie in the same direction, so there is likely to be an appreciable total error. Therefore, the figure obtained was probably low. The lift has been deduced from the induced drag, and thus there is not any experimental evidence as to the actual relationship between induced drag and lift. Works such as those of Barth or Zeder would suggest a value of $C_w = 0{\cdot}33$ as probable for this class of car, so that it can be seen that the induced drag is a prominent part of the total.

The lift generated by this particular car was considerable. This could account for reports to the effect that the car handles well, but gives the impression of being about to take off at over 75 m.p.h. At this speed, the lift is probably great enough to unload a very soft suspension, even to the extent at times of riding on the rebound buffers.

Analysis of the total resistance of the car

It is now possible to evaluate most of the factors enumerated in the resistance diagram, Fig. 2, which is the basis for the estimation of the performance of the car. Assume that the

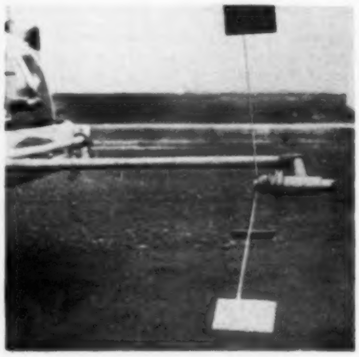


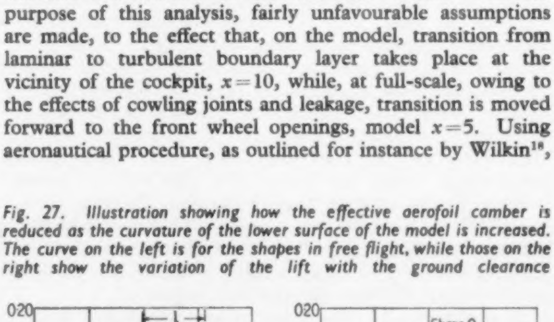
Fig. 28. Vortex circulation strength was measured by means of this free-running paddle wheel rig

relevant conditions for car 1j are as specified below:

$l = 7{\cdot}5$ ft
$2b = 5{\cdot}5$ ft
$S_w = 31{\cdot}3$ ft²
$W = 2{,}000$ lb
tyre pressure $= 35$ lb/in²
speed $= 150$ m.p.h., or $u = 220$ ft/sec in still air.

The contribution due to skin friction is liable to differ as between the model and the full-scale vehicle. For the purpose of this analysis, fairly unfavourable assumptions are made, to the effect that, on the model, transition from laminar to turbulent boundary layer takes place at the vicinity of the cockpit, $x = 10$, while, at full-scale, owing to the effects of cowling joints and leakage, transition is moved forward to the front wheel openings, model $x = 5$. Using aeronautical procedure, as outlined for instance by Wilkin[18],

Fig. 27. Illustration showing how the effective aerofoil camber is reduced as the curvature of the lower surface of the model is increased. The curve on the left is for the shapes in free flight, while those on the right show the variation of the lift with the ground clearance



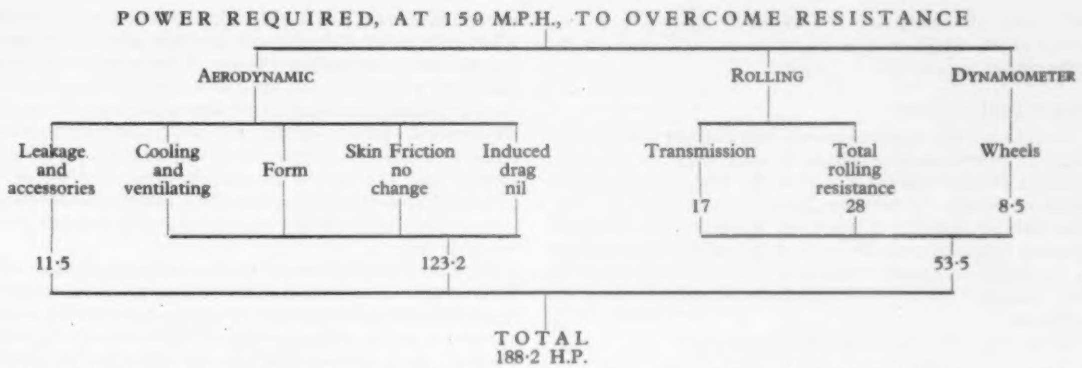


Fig. 29. Analysis of the estimated power requirement for a full-size, Grand Prix racing car developed to the stage 1j of this investigation at Ardmore

a figure for the skin friction resistance can be estimated for the two cases. In this instance, since form drag is by far the greatest part of the total, it is considered justifiable to ignore conditions in the actual transition range: this greatly simplifies the procedure. The actual skin friction coefficients, based on wetted surface area are:

in the laminar flow zone, $C_f = 1{\cdot}326\ Re^{-\frac{1}{2}}$

and, in the turbulent zone, $C_f = 0{\cdot}074\ Re^{-0{\cdot}2}$

For the model tests, the Reynolds number, based on the overall length, is:

$$Re = 2{\cdot}26 \times 10^6$$

so that, for the laminar zone, $C_{fl} = 0{\cdot}000884$

and, for the turbulent zone, $C_{ft} = 0{\cdot}00253$

The total frictional resistance thus becomes:

$$F = q\ \tfrac{1}{2}\ A_s\ C_{fl} + q\ \tfrac{1}{2}\ A_s\ C_{ft}$$

where A_s is the wetted surface area, estimated as 3 ft² for the model. The corresponding contribution to the resistance coefficient becomes:

$$\Delta C_w = \frac{q\ \frac{1}{2}\ A_s\ C_{fl} + q\ \frac{1}{2}\ A_s\ C_{ft}}{q\ S_w}$$

$$= 0{\cdot}01045$$

For the full-size vehicle at a speed of 150 m.p.h., the Reynolds Number is $Re = 2{\cdot}13 \times 10^7$, so the two skin friction coefficients are reduced slightly to:

laminar $C_{fl} = 0{\cdot}000221$

turbulent $C_{ft} = 0{\cdot}00145$

If transition has now moved forward to one-eighth of the length, then the laminar zone will cover rather less, say 0·1, of the surface area, leaving the remaining 0·9 of the area under the turbulent boundary layer. The total contribution due to skin friction then becomes:

$$C_w = \frac{0{\cdot}1\ q\ A_s\ C_{fl} + 0{\cdot}9\ q\ A_s\ C_{ft}}{q\ S_w}$$

$$= 0{\cdot}00815$$

In this case, skin friction forms about 14 per cent of the body drag, and is slightly reduced with increase in Reynolds Number. The remaining 86 per cent of the body drag is form drag, which does not alter. In the case of a special machine designed for a record attempt, it should be possible to maintain the transition zone well aft, with the result that, for the full-scale vehicle, C_w would have a value materially lower than that for the model. In the case at present under consideration, neglecting the slight reduction of skin friction, the value of C_w is 0·171, so the power lost in overcoming air resistance is:

$$P_w = \frac{u}{550}(\tfrac{1}{2}\rho\ u^2\ S_w\ C_w)$$

$$= \frac{220}{550} \times 0{\cdot}00119 \times (220)^2 \times 31{\cdot}3 \times 0{\cdot}171$$

$$= 123{\cdot}2 \text{ h.p.}$$

This figure can be considered as including cooling and ventilation losses, since these passages were open on the model. The Mercedes-Benz figure for leakage and accessory losses is $C_w = 0{\cdot}016$, which corresponds to $P = 11{\cdot}5$ h.p.

Rolling resistance of the tyres should be estimated in detail, but in the present case, all of the relevant information is not to hand. It would appear that under the prevailing conditions, a good representation of the total tyre resistance would be given by employing an effective friction coefficient of $\mu = 0{\cdot}035$. The resistance is thus $0{\cdot}035 \times 2{,}000 = 70$ lb, and the corresponding total of the power loss due to the tyres is $P_r = 28$ h.p.

Dynamometer loss is caused by the wheel, or more particularly the tyre tread, acting as a centrifugal impeller. It is uncertain how much air is really affected, but from the depth of the tyre tread it can be assumed that the affected layer on each side of the wheel is about ¼ in thick; calculations based on this thickness actually give results in relatively close agreement with the figure quoted by Romani for a typical, though rather lighter, vehicle. Test results published by Schmid also produce the same figure. Schmid went on to show that the degree of enclosure of the wheel within its cowling also has a considerable effect. In this particular case, the wheel is only partially enclosed, and the loss is estimated to be about $P_d = 8{\cdot}5$ h.p. for the four wheels. This figure is probably not greatly affected by either wheel size or tyre breadth, being mainly related to the peripheral velocity of the tyre walls, $P_d \propto \gamma^3$. The transmission loss might be generalized as 10 per cent of the total power, making in this instance $P_t = 17$ h.p. Thus, for 150 m.p.h., the total power requirement is $P_{total} = 188{\cdot}2$ h.p. These values are shown in Fig. 29. Unfortunately, there is no measured value for comparison with this total.

Cooling system

Analysis of the Ardmore model and of the closely comparable figures for the Mercedes-Benz W-196 shows that engine cooling represents only a small loss. Under certain conditions, as has been shown by Romani, it may even result in a gain. In a suitable design, the power taken by a cooling fan may be utilized so efficiently that it results in a nett gain in available thrust power, even at high speed.

Cooling of the brakes is a problem peculiar to the automobile: the brakes operate intermittently and only require cooling while in operation. On the other hand, the engine is presumably not producing power while the brakes are in operation, and therefore does not require cooling during that time. This suggests the use of a cooling system common to both engine and brakes, either with liquid circulation, as suggested by Railton, or, as has already been done by Mercedes-Benz, brakes and radiator using the same air in a common duct. The alternative is a brake cooling system that is only open to the airstream while the brakes are actually applied; this arrangement has been seen on record-breaking machines. Beyond this, little can be added to the

vast wealth of cooling data existing in the aeronautical field, nearly all of which would appear to be applicable to the automobile.

Directional stability

Model car 1 was aerodynamically unstable to a considerable degree. Reference to a previous work suggests an aerodynamic stability margin of 0·05 of the wheelbase aft of the centre of gravity. If the same figure is also accepted for the road stability margin, it would be necessary for the aerodynamic centre of pressure to be at 0·1 of the wheelbase aft of the centre of gravity, that is, at 0·6 wheelbase from the front wheels if the centre of gravity is assumed to be at the mid-point.

From Fig. 12, it was found that for car 1:

$$\frac{\partial C_s}{\partial \tau} = -0{\cdot}0166$$

and, if this side force takes effect at 0·6 wheelbase, then the resultant yawing moment about the present reference datum at the mid-point must be:

$$M\psi = -0{\cdot}0166\tau S_w ql(0{\cdot}6-0{\cdot}5)$$
$$= -0{\cdot}0166\tau S_w ql(0{\cdot}1)$$

So that $C_{M\psi} = -0{\cdot}00166\tau$

and $\frac{\partial C_{M\psi}}{\partial \tau} = -0{\cdot}00166$

The experimental value is unreliable, as there was wide scatter of the points from this particular observation, but when $\tau = 0$ it appears to be about:

$$\frac{\partial C_{M\psi}}{\partial \tau} = +0{\cdot}02$$

In other words, a considerable increase in stabilizing fin effect is required.

The investigation was continued, from this stage, using the basic solid shapes rather than detailed car models. This simplified modification, and it was felt at the same time that clearer results might be obtained. The possibility of the effectiveness of a central fin being influenced by enclosure of the cockpit at its leading edge was not appreciated at this stage.

Shape 1a was developed into shape 1b by building a longer tail to accommodate a longer and higher fin. The fin was actually made of sheet metal, while the remainder of the extra piece was built in Plasticene. This modification did give some improvement, but the yawing moment curve remained non-linear, so that the shape was decidedly unstable in the straight ahead condition. Reference to Fig. 19 shows another interesting characteristic of shape 1b. The rolling moment $C_{M\phi}$ has a high, negative slope, giving a pronounced tendency for the shape to roll towards the side wind: the centre of pressure is very low down; in fact, it is beneath the road surface. Height of the centre of pressure may have a big influence on the directional characteristics, since in general the neutral steer line is not vertical, thus the aerodynamic stability margin will be dependent on both the x and y co-ordinates of the centre of pressure. Slope of the neutral steer line simply indicates, of course, the existence of some degree of roll steer, and if the car rolls on its suspension, it takes up a slightly curved course as a result of the roll steer effect.

The fin of shape 1b was of thin section with concave sides. With a view to straightening the yawing moment curve, the sides of the fin were thickened to produce an aerofoil section, growing from the cockpit canopy. This was shape 1c, which was pleasing to the eye, and gave the effect of a streamlined body of greater length. It proved also to have the performance that would be expected of a longer and slimmer streamlined body: the non-linearity of its yawing moment curve was increased.

Straightening of the yawing moment curve was desired, because the great fin area necessary to give directional stability in the region of $\tau = 0$, with the non-linear curve, would result, apparently, in very large values of the side forces and yawing moment at large values of τ. This would probably cause great difficulty in respect of control. A rough estimate of the lateral force required of the fin showed that two, short chord, relatively high aspect ratio fins should be adequate, the total height being only that of the original fin. The central fin was then cut away and the V-tail constructed to make shape 1d.

The small change of resistance, as between shapes 1a and 1d, justifies the radial disposition of the fins. The aerodynamic directional characteristics of shape 1d are good and, furthermore, they can be altered in a predictable manner, simply by varying the length of the fins. The side force C_s was of the same order as for the long-tailed shapes and, of course, considerably higher than that of the original car. Increased side force is stated by Sawatzki to be the inevitable price of directional stability; although it is, of course, a matter of degree: the required stabilizing couple would be provided by a smaller fin at the end of a longer tail, provided that the longer body did not itself require a greater moment. At extremely high speeds, when the side force is really great, a car with large fins may be yawed by the wind to a large slip angle relative to its track on the road; this effects a balance between the lateral force of the tyres and the side force of the wind. Under such conditions, aerodynamic stabilization may not seem so attractive, but it is probably a lesser evil than instability.

At the final stage, car 1j was remarkable for the small value of all disturbing components. In fact, $C_{M\psi}$, $C_{M\phi}$, $C_{M\theta}$, and C_L are all so small that they are almost masked by the effect of experimental errors. Certainly they might all be neglected when working about the datum point used for these measurements.

So far as the application of model test results to predict the handling of the full-scale car is concerned, it is worthy of note that the actual result achieved in practice is often better than that indicated in theory. Kraus[19] has remarked that during aerodynamic development of the body for the Mercedes-Benz W-196, small stabilizing fins were found necessary to bring the centre of pressure aft to the ideal location, but that these fins were never fitted as, even without them, the full-scale cars handled very well under extreme conditions of gusty side winds.

Kraus went on to remark, with regard to wind conditions over the world's surface, that since the wind inevitably has passed over the surface of the ground for a considerable distance, the Reynolds Number and consequently the boundary layer thickness are both great. Therefore, there is a

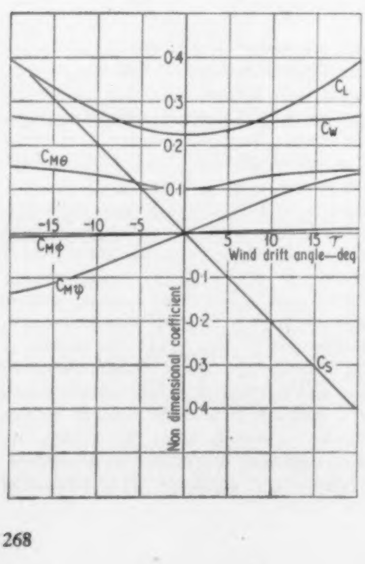


Fig. 30. A representative set of aerodynamic characteristics for a family saloon, estimated from figures published by R. Barth. The figures are based on wheelbase area, and the moments are referred to the centre of the wheelbase, at axle level

velocity gradient extending some feet above the ground. Consequently, the effective value of the drift angle, τ, for the lower part, which is of course generally the front of the moving car, is less than that for the upper part. This must exert a favourable effect on stability, but it is difficult to see how this can be turned to advantage at the design stage: before aerodynamic stability can confer any benefit, the vehicle must be directionally road stable; if the road stability margin is large, it will offset any small aerodynamic instability.

THE FUTURE

In the early days of racing, urgency of other engineering problems generally precluded aerodynamic development of the cars; furthermore, circuits were generally not such as to demand extremely high maximum speeds. In the case of the touring car, in an era when design standards gave moderate size and considerable weight, together with a normal working speed that was not high, aerodynamic treatment could give little or no benefit.

Today, racing cars are very much lighter, and circuits in general are fast, thus the significance of aerodynamic factors has increased. In fact, on occasions streamlined sports cars have proved to be faster than some of the traditional Grand Prix cars with exposed wheels. There is no doubt that the trend towards aerodynamic treatment has contributed to the remarkable increase of sports car speeds in the post-war years. It seems that a formula reducing the power available to the Grand Prix car would give some added incentive to streamlining. Since any improvement in the handling qualities of the competition machine is always welcome, it might be expected that improved streamlining will be followed by the general adoption of stabilizing fins, particularly where wind tunnel facilities are available for their development. With very light two-seater cars with small-capacity engines, a check must also be kept on the magnitude of the lift forces.

Air brakes have for a long time been used on record cars, and Mercedes-Benz have used them on their road racing cars; on a very fast circuit they would seem an attractive augmentation to the friction braking system. A brake of about 6 ft² might be accommodated at the rear of the car. This, with an effective drag coefficient of 1·0 would, for the 2,000 lb weight car already considered, give retardations as follows:

Retardation ft/sec²	Speed m.p.h.
22·0	300
9·85	200
5·53	150
2·46	100

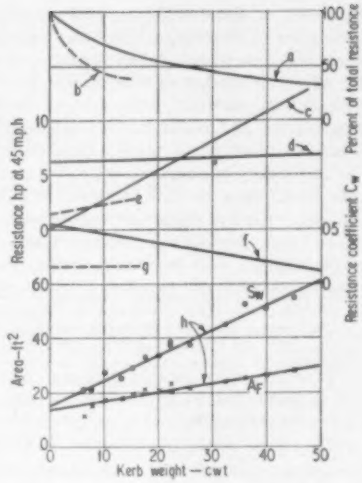
At 200 m.p.h., the air brake probably almost doubles the available retardation, while at 100 m.p.h. it is not really making a useful contribution.

The past few years have seen a remarkable increase in the performance demanded of the popular car and in the actual speeds on the road. Accommodation for the occupants, and general comfort level have also increased. Fashion has called for a general tidying up of the outside, resulting in some decrease in air resistance, while at the same time, economic pressures have led to weight reduction. Although both these factors have helped, the increased performance has been bought mainly with more power. Fig. 30, which is based on figures published by Barth, shows a representative set of aerodynamic characteristics for this class of car. Many vehicles have a high resistance and are notably unstable. Disturbing effects of side winds are common experience. By a combination of reduced size and weight, together with streamlining, the contemporary sports car reaches really high speeds, although its engine is indeed little more potent than that of the touring machine.

Fuel is already expensive, and in the long run will probably become more so. This partly accounts for the considerable interest in economy cars, which are really less than a car, but still provide tolerable personal transport. If this class of vehicle is to increase in popularity, it must have a road performance certainly not markedly inferior to that of the average car. A continuous cruising speed of 45 m.p.h. is representative of such a requirement.

Fig. 31. The effect of car size on resistance characteristics

a per cent power absorbed by aerodynamic drag; b per cent power absorbed by aerodynamic drag of a streamlined, lightweight car; c power absorbed by rolling resistance; d power absorbed by aerodynamic drag; e power absorbed by aerodynamic drag of streamlined car; f assumed trend of C_w; g resistance coefficient C_w for streamlined car; h values of S_w and A_f for fourteen different cars exhibited at the 1957 London Show



From the dimensions of some of the closed cars exhibited at the 1957 Motor Show, Fig. 31 has been constructed; the frontal area has been obtained from the general relationship $A_f = 0{\cdot}9(h-g)2b$. The scatter is surprisingly small, and the result is a clear illustration of the small decrease of frontal area as the weight of the car is reduced; at the same time, the wheel base area decreases steadily. From the aerodynamic point of view, the fineness ratio is considerably less for the lighter vehicles than the heavier ones. Consequently, the resistance coefficient tends to increase inversely as the size of car; this characteristic has been conveniently represented by a straight line for C_w, although this is an over-simplification.

In the upper part of the diagram, the effect of this trend on performance is shown. The power required to overcome the air resistance remains almost constant, while the rolling resistance decreases in proportion to the weight. At 45 m.p.h., rolling resistance is predominant for cars of 25 cwt kerb weight and upwards. For a 10 cwt car at the same speed, air resistance remains unchanged and becomes 70 per cent of the total. The dotted lines indicate what might be achieved by extreme aerodynamic treatment of the light car, even to the extent that its basic dimensions are influenced by aerodynamic considerations: at a kerb weight of 10 cwt, balance has been restored, and at 45 m.p.h., air resistance is only some 45 per cent of the total; and this total has further been reduced by some 44 per cent, which is a very useful additional saving. Such a vehicle, fitted with a transmission designed so that the engine operates at an economic speed under cruising conditions, offers a prospect of highly economical personal transport. It is probably for this class of vehicle, designed for only quite modest performance, that careful aerodynamic treatment would be most rewarding. Such a lightweight and long machine would certainly be extremely sensitive to the effects of side winds and would therefore benefit most from aerodynamic directional stabilization.

Further research

At present, it can be said that almost any required aerodynamic characteristics can be built into a car, but that this can only be done by wind tunnel development. There is sufficient published data to enable a reasonable forecast to

be made of the resistance, and perhaps even to afford an indication of the order of the other five components. Fashion must continue to govern the appearance of a product intended for sale on a market as wide as that for the popular car, even though the indirect influence of such considerations as engineering and aerodynamics can ultimately result in great changes. Because of market requirements, there will never be uniformity of shape, so tests of groups of idealized bodies can be of only limited interest. Wind tunnel testing may become a normal design procedure with a view to controlling noise and ventilation rather than obtaining increased performance. It is hoped that many more wind tunnel test figures for cars and car-shape bodies will be published, as it is only from their study and comparison that a background of understanding can be built up as a foundation for future designs.

There remains the disparity between model and full-scale tests, which has been recorded by Romani. The investigation of under-surface roughness, shown in Fig. 22, might be a clue to the origin of the effect; leaks and, in particular, air discharges beneath a real car could have a greater influence than the solid bars used in the experiment, in which the effect on lift and pitch was found to be far more pronounced than that on drag. This matter can only be resolved by road tests of a car that is truly aerodynamically similar to its wind tunnel model; both model and car must have all leaks carefully eliminated. Wind tunnel test of the full-scale car would be of even greater interest. On the road test, lift could be assessed by a comparison of surface pressures with those of the scale model, and a cross-check made by recording the deflections of the road springs and using them as a measure of the load supported, and hence the lift.

The production of streamline diagrams, by one of the flow visualization techniques, showing the airflow pattern in the vicinity of a few different body shapes would be a great help to the designer. Also, a theoretical treatment of the streamline pattern associated with the complete car and mirror-image, indicating the lift and drag effects, would certainly be valuable. Some more tangible reports on road tests of aerodynamically stabilized cars would be of great interest; the German works, such as those of Huber[20], refer to tests, but appear to give few details.

Related to this full-scale work is the matter of vertical wind-gradient. If some records were kept, a statistical analysis might be compiled to afford a more accurate basis for design and for wind tunnel testing. Although not primarily an aerodynamic problem, the analysis of losses associated with the tyres at high speeds, and in particular the dynamometer loss, appears to be still incomplete. The complexities of compressible flow dynamics are unlikely to affect the automobile field, at least, not in the forseeable future.

References

13. C. EARL: "Investigation into the Development of German Grand Prix Cars Between 1934 and 1939 (Including a Description of the Mercedes World's Land Speed Record Contender)", *B.I.O.S. Final Report No.* 1755, *Item No.* 19, H.M. Stationery Office.
14. R. BARTH: "Einfluss der Form und der Unstromung von Kraftfahrzeugen auf Widerstand, Bodenhaftung und Fahrtrichtungshaltung", *Z.VDI*, Vol. 98, No. 22, 1st August, 1956.
15. R. BARTH: "The Aerodynamics of Car Body Shapes", *Engineers' Digest*, Vol. XVII, No 10, October, 1956. (This is a condensed version, in English, of Reference 14.)
16. T. VON KARMAN: "General Aerodynamic Theory—Perfect Fluids", *Guggenheim Trust*, and in "Aerodynamic Theory", Vol. II, Division E, edited by W. F. Durand, *California Institute of Technology*, 1934.
17. C. F. TOMS: "Introduction to Aeronautics", Griffen and Co., 1947.
18. E. W. C. WILKINS: "Aeroplane Design", Griffen and Co., 1938.
19. L. KRAUS: "Rennfahrzeuge", *VDI Berichte*, Band 16, 1956.
20. L. HUBER: "Die Fahrtrichtungsstabilitat des Schnellfahrenden Kraftwagens", *Deutsche Kraftfahrtforschung*, Vol. 44, No. 1, VDI, 1940.
21. G. E. LIND WALKER: "Directional Stability", *Automobile Engineer*, Vol. XL, August and September, 1950.

ROAD SPEED TYRE

A NEW version of the Dunlop Road Speed tyre has been introduced. It has a strong, all-nylon casing and a new synthetic rubber tread. This tyre, which is known as the R.S.4, has been developed by the Dunlop Rubber Co. Ltd. for application to production cars designed for high speeds. It has been well tested and operated on cars driven at very high speeds on Continental motorways.

The nylon construction gives maximum safety, since it greatly increases resistance to all kinds of impact and reduces the danger of bursts or tyre failures at high speeds. Also, it is more resistant to damage from heat generated internally by rapid flexing of the tyre at high speeds of operation. Road holding is said to be good even on wet or slippery surfaces; this feature is obtained by virtue of the pattern of the tread and the nature of the synthetic rubber compound used. Dunlop estimate that in wet conditions, the grip obtained with the synthetic rubber approaches a figure of 30 per cent better than that obtained with other rubber compounds used hitherto. The tread also has been designed with a view to providing maximum resistance to wheel spin and therefore to give effective torque transmission.

Other advantages claimed are quiet running, improved resistance to tyre squeal, and longer tread life. At present, the tyre is made in the following sizes: 5.50—15; 5.90—15; 6.00—15; 6.40—15; 6.00—16; 6.50—16; 6.70—16; and all can be supplied with black or white side walls.

METAL SPRAYING CONFERENCE

IN view of the great success of the first International Metal Spraying Conference held in Halle, East Germany, in 1956, it has been decided to hold a second one, this time in Great Britain. The second conference will be in Birmingham, from the 29th September to the 3rd October, 1958, and all the arrangements are being undertaken by the Association of Metal Sprayers. Lectures and discussions will take place at the new College of Technology, Gosta Green, Birmingham, and the programme includes practical demonstrations and works visits.

A large number of papers have been submitted by experts from all over the world. They include the following subjects: practical developments and applications in various countries and climates; industrial hazards; building up and hard surfacing; diffused coatings; use of high melting point metals; mechanical and chemical properties of deposits; linings for anti-friction bearings; electric arc spraying; and, deposition of plastics and refractories by fusion spraying. The proceedings will be translated simultaneously into French, German and English, and transmitted by means of an internal radio telephone installation. Programmes and forms of application for registration can be obtained from The Association of Metal Sprayers, the address of which is Barclays Bank Chambers, Dudley, Worcestershire, England.

GAS EJECTORS*

Theoretical Considerations Relating to Testing and Performance

Cand. Sc. Tech. A. G. FILIMONOV

FOR the purpose of this article, an ejector is defined as an apparatus by means of which the total pressure of a gas stream is increased at the expense of another stream of gas at higher pressure. The basic scheme of such an apparatus is illustrated in Fig. 1, in which gas stream G_1, emerging from the nozzle *cb*, moves the air from the inlet chamber, through the nozzle *ck*. It does this by the turbulent mixing of the ejecting and ejected flows in the mixing chamber. In the diffuser, the velocity head of the combined stream is partly transformed into a pressure head. The flow of gases out of the entry chamber gives rise to a depression in this chamber, so more air, represented by the flow G_2, is drawn in.

Hitherto, little use has been made of gas ejectors in automobile engineering and design. Nevertheless, despite the fact that there are many practical problems involved, their application is worthy of serious consideration because of their basic simplicity. For example, an ejector might be used instead of a cooling fan and drive assembly. In this application, exhaust gas energy could be employed as the activating force; this would have the advantage that the energy available would to some extent vary appropriately with the cooling requirements under different conditions of load. A system of this type might be used in conjunction with an oil cooler. Another possible application is automatic cleaning of dry filters for engine air intakes. Also, an ejector could be used to reduce the exhaust back pressure when the vehicle is travelling at speeds in excess of about 40 m.p.h. In the latter case, the energy of the air stream flowing past the vehicle could be used to eject the exhaust gases.

Work done by the ejector is determined by the weight pressure and temperature of the ejecting, or primary, flow, and ejected, or secondary flow. These parameters are usually expressed in terms of non-dimensional ratios:

$$\frac{h_2}{h_1}=f(q)$$

and

$$q=\frac{G_2}{G_1}$$

where h_1 is the excess static pressure upstream of the primary flow nozzle; h_2 the depression upstream of the secondary flow nozzle, that is, at the entry chamber; and G_1 and G_2 are the weights of the primary and secondary flows respectively.

Experimental investigation of gas ejectors, as discussed in this article, is based on the control of pressure by regulation of the volume of gas passing from the compressors through the nozzles. The excess pressure supplied by the compressors compensates for the aerodynamic losses due to the flow measuring equipment. This ensures reliability and adequate accuracy over the entire test range, so far as the determination of G_1 and G_2 is concerned, regardless of whether the flows are of the steady state or pulsating types.

Fig. 2 shows the layout of the test rig. The primary flow is supplied through a flow volume measuring device 2, pipe 3 and a piston pulsator 4, to the pipe 1. At the same time, the secondary flow is admitted through the flow measuring device 6 to the entry chamber 5. The surge tanks 7 and 8 smooth out pulsations due to the action of the flow measuring devices 6 and 2 respectively. Gauges 9 and 10 and thermometers 11 and 12 serve to determine the gas conditions

cb high pressure, or primary, gas nozzle; **ck** *low pressure, or secondary, gas nozzle;* **kb** *inlet chamber;* **kc** *mixing chamber;* **dφ** *duffuser; 1, 2,* **a, b** *and* **c** *the cross sections*

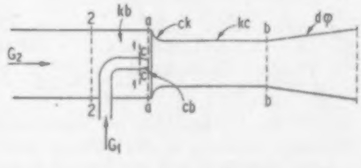


Fig. 1. Basic principle of a gas ejector

upstream of the flow measuring devices. Static pressures h_1 and h_2 are measured by the gauge 13 and micromanometer 14, respectively. Nozzles 15 and 16, as well as the mixing chamber 17 and diffuser 18, can be replaced by units of different size. The flows are controlled by valves 19 and 20.

It is assumed, to start with, that at subcritical velocities $p_c=p_a$ and with the ejector working under steady state conditions, the equation of continuity of momentum at sections *a-a*, *c-c*, and *b-b*, Fig. 1, applies as follows:

*Translated from "Avtomobilnaia Promishlenost," February, 1958.

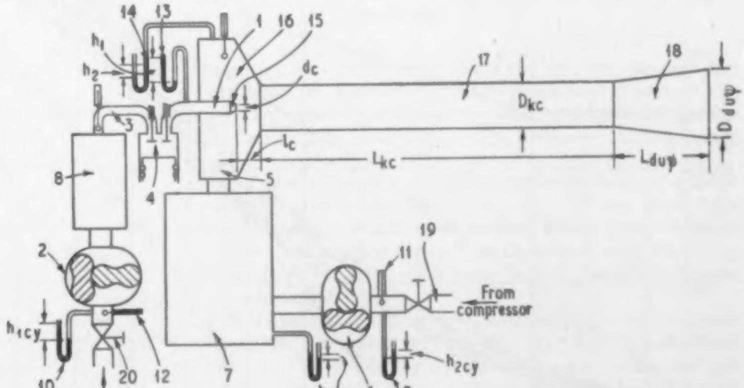


Fig. 2. This illustration shows the layout of a rig for the experimental testing of ejectors

$$W_{dc}\, dM_c + \cos\alpha\, W_{da}\, dM_a - W_{db}\, dM_b = p_b df_b - p_a d(f_a + f_c) + p_a d(f_a + f_c - f_b)$$

where p is the static pressure, kg/m²; W_d the true gas velocity, m/sec; α the angle between the direction of W_{da} and W_{dc}; M the gas flow per second, kg-sec²/m-sec, and $dM = W_d \gamma df/g$; γ the specific weight of gas, kg/m³; and f the area of cross section, m². The indices c, a and b refer to the cross section of the primary flow nozzle, the area of flow of the secondary stream at the face of the primary nozzle, and at the exit from the mixing chamber, respectively, Fig. 1.

Assume γ_c, γ_a, γ_b, p_a and p_b as constant at the relevant cross sections, and replace W_{dc}, W_{da} and W_{db} by mean values with a coefficient of unsteady distribution; then:

$$\frac{k_c W_c^2 \gamma_c f_c}{g} + k_a \cos\alpha \frac{W_a^2 \gamma_a f_a}{g} - \frac{k_b W_b^2 \gamma_b f_b}{g} = (p_b - p_a) f_b$$

where $k = (1/fW^2) \int_0^f W^2{}_d df$ is the coefficient of uneven distribution of the flow velocity over the cross section; W is the mean gas velocity, m/sec; and $G = W\gamma f$.

On the other hand:

$$p_a = p_2 + \frac{\gamma_2 W_2^2}{2g} - \frac{\gamma_a W_a^2}{2g} - \xi_{bx} \frac{\gamma_a W_a^2}{2g} = p_2 + \frac{G_2^2}{2g\gamma_2 f_2^2} - \frac{G_2^2}{2g\gamma_a f_a^2}(1 + \xi_{bx})$$

so that:

$$p_b - p_2 = \frac{k_c G_1^2}{g\gamma_c f_c f_b} + \frac{k_a G_2^2}{g\gamma_a f_a f_b}\cos\alpha + \frac{G_2^2}{2g\gamma_2 f_2^2} - \frac{G_2^2}{2g\gamma_a f_a^2}(1 + \xi_{bx}) - \frac{k_b (G_1 + G_2)^2}{g\gamma_b f_b^2}$$

Allowing for a gain in pressure in the diffuser, and friction losses in the ejector proper:

$$p_0 - p_2 = \frac{k_c G_1^2}{g\gamma_c f_c f_b} + \frac{k_a G_2^2}{g\gamma_a f_a f_b}\cos\alpha + \frac{G_2^2}{2g\gamma_2 f_2^2} - \frac{G_2^2}{2g\gamma_a f_a^2}(1 + \xi_{bx}) - \frac{k_b (G_1 + G_2)^2}{g\gamma_b f_b^2} + \frac{k_b (G_1 + G_2)^2}{2g\gamma_b f_b^2}\psi d - \frac{k_b (G_1 + G_2)^2}{2g\gamma_b f_b^2}\xi mp \quad \text{(a)}$$

where ψ_d, ξ_{bx} and ξ_{mp} are respectively the coefficients of pressure recovery in the diffuser and at the entry into the mixing chamber, and the friction losses in the ejector. The suffixes 1 and 2 refer to the cross sections upstream of the primary and secondary flow nozzles respectively, Fig. 1.

To obtain the basic equation of the ejector, it is necessary to divide both parts of equation (a) by the excess static pressure upstream of the nozzle for the primary flow h_1; this excess pressure is determined from the equation of flow through the nozzle, making the appropriate allowance for the starting velocity, and from the equation for flow continuity.

The pressure drop in the ejector nozzle is:

$$\Delta h = h_1 - h_c = \frac{G_1^2}{2g\gamma_c \mu^2 f_c^2}\left[1 - \mu^2 \left(\frac{d_c}{d_1}\right)^4 \left(\frac{\gamma_c}{\gamma_1}\right)^2\right] \quad \text{(b)}$$

where μ is the coefficient of consumption, h_c is the excess static pressure at the exit from the primary flow nozzle; and d_c and d_1 the diameters of the nozzle and duct respectively upstream of the nozzle.

When the gas is discharged to atmosphere, the pressure at the exit from the nozzle is equal to that of the atmosphere, that is $h_c = 0$ and $\Delta h = h_1$. If the ejector works at $G_2 = 0$ at the cross section of the nozzle, the depression will be equal to that maintained at the entry chamber, that is, $h_c = h_2$ and $\Delta h = h_1 - h_2$. The magnitude of $\Delta h = h_1 - h_2$ depends only on the value of G_1 and the nozzle parameters, and is unaffected by the characteristics of the flow passages of the ejector.

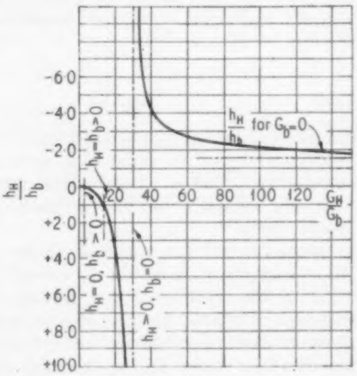


Fig. 4. Nozzle characteristics in terms of the non-dimensional co-ordinates G_H/G_b and h_H/h_b

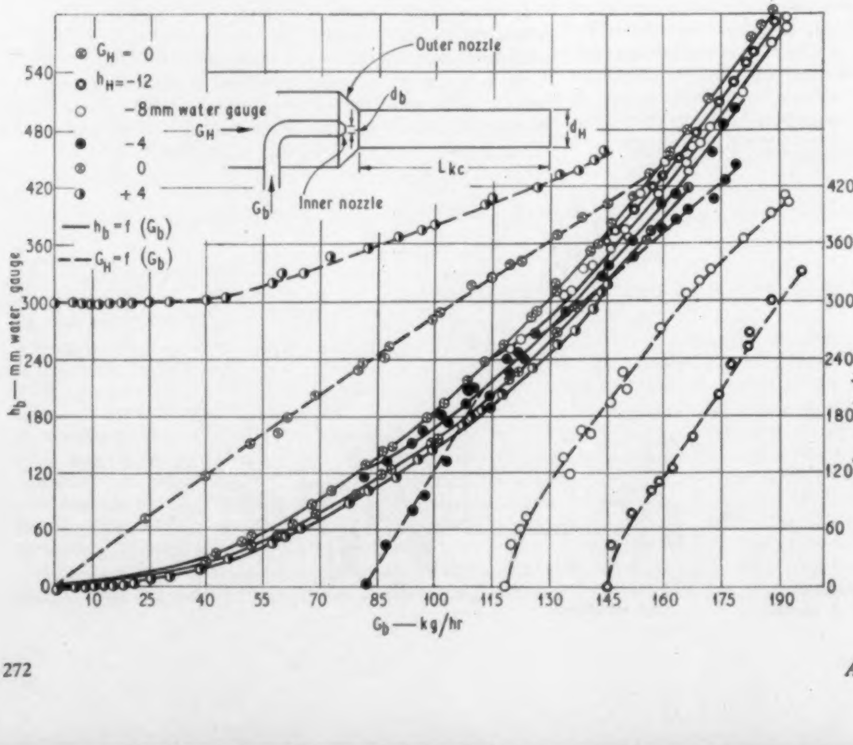


Fig. 3. Results of test carried out to determine nozzle performance, where d_b = 25 mm, d_H = 200 mm, L_{kc} = 1,500 mm

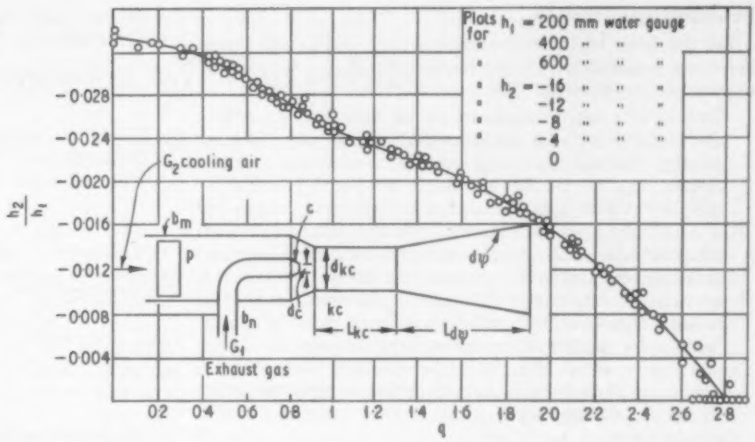


Fig. 5. Basic layout and the non-dimensional characteristic of an engine-cooling ejector

p radiator; b_m cooling-air duct; b_n exhaust gas entry; d_c exhaust gas nozzle; kc mixing chamber; $d\psi$ diffuser

A secondary flow G_2 cancels the singularity of the function $h_1-h_2=f(G_1)$. So long as the primary flow remains constant, an increase in G_2 will reduce h_1. This can only be accounted for by a change of pressure at the nozzle and of Δh, that is, of h_1-h_c. From this:

$$h_1-h_c=(h_1-h_2)_{q=0} \tag{c}$$

and $G_1=idem.$

Solving simultaneously equations (b) and (c):

$$h_1=\frac{G_1^2}{2g\gamma_c\mu_c^2f_c^2}\left[1-\mu_c^2\left(\frac{d_c}{d_1}\right)^4\left(\frac{\gamma_c}{\gamma_1}\right)^2\right] \tag{1}$$

where:

$$\mu_c=\mu\sqrt{\frac{W_1^2\gamma_c+2g\,(h_1-h_c)}{W_1^2\gamma_c+2gh_1}}$$

Comparing the coefficient of consumption μ with the coefficient μ_c, it can be seen that the latter takes into account the effects, on the static pressure upstream of the nozzle, of both the secondary air flow and the pressure drop at the intake chamber.

By dividing the left- and right-hand sides of equation (a) by the corresponding parts of equation (1), and rearranging the results, it is possible to obtain the basic equation for an ejector operating with steady state flows:

$$\frac{h_2}{h_1}=\frac{2k_c\psi_c^2}{am}+\frac{\mu_c^2q^2}{a\Delta n}\left(\frac{2k_a\cos\alpha}{m}-\frac{1+\xi_{bx}}{n}\right)+\frac{\mu_c^2q^2}{a\Delta_1m_1^2}-\frac{2k_b\mu_c^2(1+q)(1+\frac{q}{\Delta})(2-\psi_d+\xi_{mp})}{am^2} \tag{2}$$

where:

$$\psi_c=\psi\sqrt{\frac{W_1^2\gamma_c+2g(h_1-h_c)}{W_1^2\gamma_c+2gh_1}}$$

while ψ is the velocity coefficient

$$a=1-\mu_c^2\left(\frac{d_c}{d_1}\right)^4\left(\frac{\gamma_c}{\gamma_1}\right)^2$$

$m=\frac{f_b}{f_c}$ a scale coefficient

$$m_1=\frac{f_2}{f_c}$$

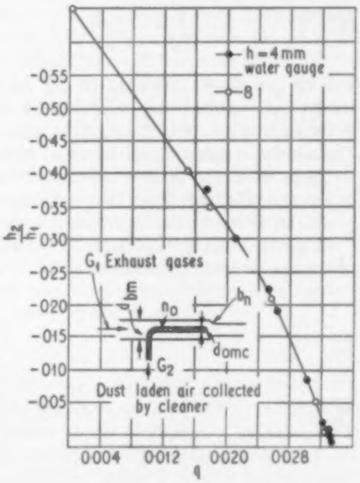
$$n=\frac{f_a}{f_c}$$

$$\Delta=\frac{\gamma_a}{\gamma_c}=\frac{T_1}{T_2}$$

when assuming $p_c=p_a$

$$\Delta_1=\frac{\gamma_2}{\gamma_c}$$

From the foregoing discussion, it can be seen that the gas ejector is simply a system of nozzles; in the simplest case, there are only two nozzles, one inside the other. The performance of such a system, with which the main parameters are altered within a wide range, will now be considered.



b_n exhaust duct; n_0 dust-removing duct

Fig. 6. Fundamental scheme and non-dimensional characteristic of an ejector for servicing an engine air cleaner

Results of tests on an ejector with circular nozzles are plotted in Fig. 3. The diameters of the primary and secondary nozzles used in these tests are 25 mm and 200 mm respectively, and the length of the mixing chamber is 1,500 mm. All the results are plotted on a non-dimensional basis, in terms of:

$$\frac{h_H}{h_b}=f\left(\frac{G_H}{G_b}\right)$$

where G_H, G_b, and h_H and h_b are the flow weights and the static pressures respectively of the gases emerging from the inner, or primary nozzle, and the outer, or secondary, nozzle.

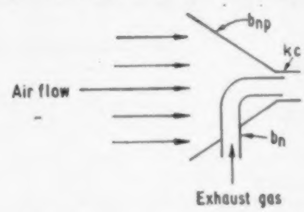
It can be seen from Fig. 4 that there are two operating regions of the nozzles. They are:

1. The operating range obtainable with a centrally-positioned high pressure gas supply, $h_b>h_H$. If the previously utilized designations, with indices 1 and 2, are applied to the flows and pressures, then $G_1=G_b$, $h_1=h_b$, $G_2=G_H$, and $h_2=h_H$, and so the basic ejector equation (2) can be employed.
2. That obtainable with a peripheral supply of high pressure gas. In this instance, $G_1=G_H$, $h_1=h_H$, $G_2=G_b$, and $h_2=h_b$. To determine the basic ejector equation for this case, divide both parts of equation (a) by h_1, calculated on the basis of gas flow through the outer nozzle.

Practical applications

On the basis of the relationship $h_H/h_b=f(G_H/G_b)$, there are three possibilities for the application of gas ejectors in automobile engineering. These are as follows:

1. Instead of a fan, to maintain the air flow through either the radiator or, with an air-cooled engine, the cylinder finning. Exhaust gas energy could be used for the ejection nozzle, Fig. 5. By this means, it would be possible to simplify engine design, increase the effective output by 5 to 10 per cent and correspondingly reduce the fuel consumption. The ejectors would have a central primary flow supply, ratios of secondary : primary nozzle areas of between 10 : 1 and 25 : 1, or even more, and long mixing chambers and diffusers. These ejectors would have to be capable of passing a large volume of cooling air, that is, about 35 to 50 in³ per effective horse power, against a relatively low head, which would be of the order of 30 to 60 mm water gauge. Their working range would be:
$$0<\frac{G_H}{G_b}<\left(\frac{G_H}{G_b}\right)_{h_H=0}$$
2. For automatic servicing of the dust collecting bowls of dry air-cleaners, for example, of the centrifugal or felt types. This would simplify maintenance, as well as increase engine life. In this application, the exhaust gases would be fed into the ejector through the peripheral nozzle, Fig. 6. The best arrangement is generally for the exhaust duct to serve as the mixing chamber, and to dispense with the diffuser. Ejectors of this type produce an appreciable pressure drop with small secondary flows in the dust removing duct. Their working range is determined by:
$$\frac{G_H}{G_b}>\left(\frac{G_H}{G_b}\right)_{h_b=0}$$
3. To reduce exhaust back pressure by using the energy of the air stream, Fig. 7, at speeds higher than 40-50 km/hr. This improves cylinder scavenging, and thus increases the weight of the air charge and improves fuel consumption. For this application, the ejectors usually have a rather short mixing chamber, without a diffuser, and the ejector works under conditions defined by:
$$\frac{G_H}{G_b}>\left(\frac{G_H}{G_b}\right)_{h_H=0}$$



bn *exhaust duct;* **bnp** *air entry funnel;* **kc** *mixing chamber*

Fig. 7. Basic arrangement of an ejector for reducing exhaust back pressure

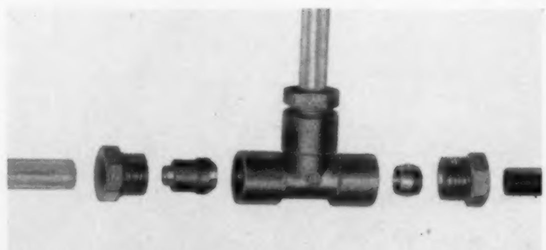
The data plotted in terms of dimensionless coordinates, q and h_2/h_1, in Figs. 5 and 6, for cooling system and air cleaner ejectors, were obtained by determining experimentally the performance of units with nozzle dimensions $d_b=25$ mm, $d_H=200$ mm and $L_{kc}=1{,}500$ mm, and with the ejector working under steady state conditions of engine operation.

TUBE JOINTS

NYLON tubing is now being employed in an ever-increasing variety of applications in this country, and has been well tried and proved for several years in the United States of America. It can be used for hydraulic, pneumatic, lubrication and fuel systems. This material is used not only as an alternative to copper and steel, but also for applications for which metal tubing is unsuitable.

To meet adequately the problem of satisfactorily jointing nylon tubing, Compression Joints, Ltd., of Tyburn Road, Birmingham, 24, have designed and patented a nut type compression fitting. This component is termed the Nitex fitting and is manufactured from high quality brass. A special feature of the design is that it provides for controlled compression to ensure that the tube is not damaged when the joint is tightened. The joint can be made, broken and remade any number of times without losing its efficiency, and special tools are not required. The Nitex ferrule does not distort: this is claimed to be an improvement on most metallic compression joints. The ferrule incorporates a tube liner that prevents the collapse of the tube under compression. Nitex fittings are equally suitable for use with copper tubes of small diameters; for these applications, conventional tube nuts and olives can be supplied. A wide range of Nitex fittings will shortly be available for nylon and copper tubes of the following sizes: $\frac{1}{8}$ in, $\frac{5}{32}$ in, $\frac{3}{16}$ in, $\frac{1}{4}$ in, $\frac{5}{16}$ in, and $\frac{3}{8}$ in outside diameters.

A Nitex T-junction. On the left are shown the components for use with nylon tubes, and on the right, those for use with metal tubes



WHITEHEART MALLEABLE IRON CASTINGS

TWO grades of whiteheart malleable iron castings, W22/4 and W24/8, form the subject of a revision of the 1947 edition of B.S. 309. The chemical composition of the metal of the castings is considered to be of less importance than the mechanical properties; in consequence, compliance with the requirements of the specified tests is to be regarded as the main criterion of the material's suitability for purpose.

The provision and heat treatment of two sizes of test bar are specified, as also are tensile and bend tests; provision is made for additional tests to be undertaken by arrangement with the manufacturer. Alternative dimensions given for the tensile test bars are based on practice in several other countries. They have been included in order that experience with them can be gained in the United Kingdom, and to facilitate the exchange of information with other countries. Copies of this Standard, B.S. 309 : 1958, can be obtained from the British Standards Institution, Sales Branch, 2 Park Street, London, W.1. The price is 4s., plus postage.

Strength of Materials

Fundamental Research Required in a Wider Field than Hitherto

IN a paper entitled "On the Necessity of Fundamental and Widened Research in Strength of Materials," published in the December 1957 issue of the *ASTM Bulletin*, the author, M. V. Zaustin, outlines a number of new design requirements that have arisen in recent years with regard to data on properties of materials. Among these are the following. Requirements with regard to minimum weight and maximum strength are now more rigorous than hitherto—incidentally, there is a strong case for more detailed studies of the relationship between strength, cost and weight. Machines, including internal combustion engines, are operating at higher speeds than ever before, and the fact that vehicles travel faster leads to higher stresses, fatigue loading and mechanical shock loading. Also, better quality materials are now available and mechanical test results on them give a relatively small scatter; consequently, there is an unexplored possibility of using smaller factors of safety.

All these features of modern engineering require not only more accurate design calculation, but also correspondingly more accurate and more suitable test data. This will enable the design engineer to predict with confidence the behaviour and strength of his designs. The gap between the strength of materials, as determined in laboratories, and the actual strength of components can no longer be bridged by large factors of safety.

The fields of stresses and strength of materials must be brought closer together. What separates these two fields? In most cases, the discrepancy between the strength of the design elements and the strength of the specimens made of the same material is caused by the effect of the stress system, or pattern, on the strength of materials. Regardless of whether a given stress combination is obvious, as in pressure vessels, or is hidden, as in cases of stress concentrations and residual stresses, its effect is always present. The study of the relationship between the strength of the design element in service and the strength of materials, measured in laboratories, should become a special branch of materials engineering, which, when applied to metals, might be called *metal mechanics*.

Such study has little to do with the metallurgy that deals with the chemical and structural transformations of metals and alloys, nor with the strength of materials, which is essentially a science of transmission of forces. Rather is it the child of the art and science of testing materials. Its sole purpose is the scientific determination of the allowable stresses. This translation of the strength of specimens in terms of the strength of structural elements is the greatest problem of modern experimental and theoretical strength of metals.

On one hand, there are the results of laboratory tests of materials in which a straight, smooth specimen is gradually loaded, the stress distribution is uniform, and the temperature is constant, under standard laboratory conditions. While on the other hand, the design element is not straight, is not always smooth, and generally has holes and cut-outs; its stresses are not uniformly distributed and are not simple, but are combined; the load is suddenly increasing or decreasing, the temperature is changing, and the surrounding medium is not neutral. Therefore, the standard experimental data cannot necessarily be applied directly to a structural member, although there are cases in which ordinary laboratory data can be directly applied. The gap between specimen test data and results obtained in practice is increased considerably by the new factors, mentioned in the first paragraph, and the old method of closing this gap by means of increased factors of safety has become both dangerous and expensive.

This is where the science of metal mechanics could be of use. It takes into account all those features and minor factors that differentiate a design element from a test specimen. In reality, these factors are not minor, so far as their practical effects are concerned, for it may happen that while a specimen made of material A is stronger than the identical specimen made of material B, an actual part made of A is weaker than the identical part made of material B.

The purpose of metal mechanics is not only to explain such facts, but also to supply data that will permit the designer to make a correct estimate of the real strength of design parts and thus avoid needlessly large or dangerously small factors of safety. The ideal of modern design engineering is to leave to the margin of safety only those factors of strength which by their nature belong to the field of probability. Such factors include the non-uniformity of material, the uncertainties of production, and unexpected behaviour of loading. But even these factors may be treated with mathematical exactness by the theory of probability.

Theory of elasticity

While materials data are based on idealized and standardized test conditions, stress analysis also suffers from simplifications and assumptions that are often far from realistic. The theory of elasticity deals with deformations and stresses in the elastic range, but entirely ignores the fact that most failures occur in the plastic range. For this reason, the stresses calculated from ordinary formulae are often not the final design stresses, since they represent the actual conditions only in the simplest cases, and often must be altered to satisfy the existing conditions. Here is where the role of metal mechanics would serve as a connecting link between the strength of a part and the strength of a metal.

Modern engineering cannot be satisfied with the fundamental assumption of isotropy or perfect elasticity. It seeks to take into account factors such as the directionality of mechanical properties of materials. Different modulii of elasticity are operative in different directions, that is, transverse and longitudinal, tensile and compressive. In some other cases of precision design, Hooke's relationship is rejected and replaced by stress-strain curves, because in many instances, for example aluminium alloys and stainless steels, the proportionality relationship between stresses and strains is only approximate. In such cases, tangent, secant, and reduced modulii have to be used. There are other circumstances in which the classical theory of elasticity should be modified to take into account anelastic effects that cause the deformation to be a non linear function of applied stresses. An example of this is a vibrating body with internal friction, when energy must be supplied to maintain vibrations at a constant amplitude. From the foregoing discussion, it is evident that there is a need for comprehensive stress-strain data, not only in the elastic range but in the plastic range, and not only for tension but also for compression.

When stresses are not uniformly distributed, the theory of elasticity again comes into conflict with reality, owing to the

fact that yield and ultimate strengths belong to the domain of plastic deformations. If, for instance, a beam is subjected to bending by a moment that is calculated from the ordinary bending formula, using a value for the yield strength based on 0·2 per cent permanent elongation, the permanent elongation of the outermost fibre will be less than 0·2 per cent, probably 0·15 per cent. To obtain a permanent deformation of 0·2 per cent, the load will have to be increased, sometimes by 20 per cent or more. The explanation is simple: the inner fibres of the beam are stretched elastically and pull back the outer fibres, which are stretched plastically; this reduces the permanent set. A similar situation is observed when components are stressed torsionally, or in any other cases where a stress gradient is present.

This is an important situation, because one of the most widely used principles of rational design is the employment of the same criterion of strength for all design elements. In this case, the same permanent deformation that is used for yield strength should be used for bending. Adherence to this principle may result in a considerable saving of weight. Since, in bending and torsion the magnitude of the residual deformation depends on the relationship between the elastic and plastic properties of a material, and since this relationship cannot be derived from ordinary tensile data, the design engineer must have special experimental data, for bending and torsion, in the form of moment-deformation or stress-strain curves.

The theory of elasticity fails to answer many questions. Why does an axial force, applied to a specimen in tension, produce a component that is perpendicular to the axis, causing the Poisson's effect? Why does a cube made of brittle material, subjected to compression, fracture in tension? Or, by what right are the results of a tension test of a metal that forms a neck before breaking applied to the calculations of strength of a pressure vessel, which breaks without forming a neck or a noticeable plastic deformation? By what right is the yield or ultimate tensile strength, which belongs to the plastic range, used in formulae for bending and torsion, which are derived for the elastic range? Or, why are there so many theories of strength, none of which gives a reliable answer?

The use of the strongest material is often far from being the correct solution in designing the strongest part. In selecting material for a pressurized tank, an inexperienced designer may wish to take advantage of the mechanical properties of a steel heat treated to a high strength, without realizing that a greater ductility of the same material, treated to a lower strength, will more than compensate him with respect to strength and safety. Again, selection of a half-hard material because of its greater degree of straightness, for a strut in compression often results in a stronger strut than would use of a full-hard material. Where there are stress concentrations, components made of stronger metals are often weaker than identical parts made of weaker but more ductile materials. With steels of medium strength, of the order of 120,000 lb/in^2 ultimate tensile strength, fatigue limits are often obtained that are almost equal, and sometimes even superior to, those of steels of practically twice the static strength. There are many other factors that arise where the accuracy of strength prediction is a primary factor in design for minimum weight and maximum safety.

The problem with regard to the presentation of data on materials in such a way that it can be directly applied in engineering design is basically that the materials engineer does not know what properties to measure. The data that have been obtained in the past are of most value as a measure of the relative worth of materials and are not usually suitable for direct application by the designer.

In order to deal successfully with the above-mentioned and similar problems, the nature of the strength of materials, which is a combination of properties of materials and the properties of stresses, has to be understood. Only deeper detailed study of the relationship between the properties of materials and the properties of stresses will explain many phenomena and answer many questions hitherto unanswered, as well as enable us to obtain more realistic data on the mechanical properties of materials.

Not so long ago it was believed that a material always possesses the same mechanical properties, regardless of the kind of forces applied and regardless of the character of stresses. Now it is known that brittleness and ductility, for instance, are not inherent properties, but only mechanical states of a metal. It is known that the character of stresses and the mode of their application may change the properties of a material even to a greater degree than do chemical composition, heat treatment, or temperature.

Detail considerations

In his paper, the author then goes on to discuss in detail various aspects of the problem. For example, he states that the character of a fracture does not depend on the absolute values of stresses and strengths, but on their ratios. This conclusion fundamentally changes the old concept that each material has only one mode of fracture. It may be safely stated that the adherence to this one-sided idea retarded the development of a rational, unified theory of strength. It was responsible for the split of all theories of strength into two groups, without any indication of which group should be used in any given case.

Another important conclusion is that a brittle fracture may occur as result of several causes. Since shear resistance is sensitive to all conditions of tests, we may expect a brittle fracture under any one of those conditions that increase the shear resistance and make it greater than the cohesive resistance. This may happen under the following conditions:

Low temperature.
High speed of deformation.
Change in crystalline state, such as is caused by heat treatment or strain hardening.

On the other hand, brittle fracture may be caused by a reduction of the tangential stress to such an extent that the cohesive strength is overcome first by the normal stress. This might happen under biaxial or triaxial stress conditions.

A question arises as to whether cohesive strength is a really important property of metals? Since it has been possible to manage to get along for a long time without this data, why is it now necessary? In the first place, it is impossible to predict brittleness or ductility of the design elements under a full load, especially with combined stresses, without knowing the value of cohesive and shear strengths. Such prediction becomes very important when factors of safety are small.

In addition to the combined stresses produced by the external forces, there are equally dangerous stresses of internal origin, that is, residual stresses, the stresses produced by stress raisers and probably fatigue loading. It is well known that the first two kinds of stresses are biaxial and triaxial stresses, which are responsible for a brittle fracture. It was recently found that fatigue loading reduces cohesive strength; and, therefore, high cohesive resistance is desirable for fatigue resistance.

Another section of the paper is devoted to suggestions for the improvement of methods of testing materials. In it, the author discusses testing in relation to tension, compression, torsion and shear. He also covers the subject of stress concentrations and the need for data concerning the properties of materials stressed beyond the elastic range. Impact and fatigue, the effects of high and low temperatures are dealt with, and a case is stated demonstrating the need for probability distribution data concerning the strength of materials, because minimum strength values are not absolute.

Vauxhall Motors' New Press Shop

This Plant, a Major Feature of the Vast Expansion Programme at Luton, Embodies the Most Advanced Layout and Practice. It is arranged for Cyclic Production

PROBABLY the most adequate impression of the size and potentialities of this plant will be conveyed by the statement of a few relevant facts and figures. The press shop is 882 ft long and 400 ft wide. Running alongside for the full length, and separated only by a wide gangway, is the 100 ft wide steel store. The main area is divided transversely into eleven bays, nine of which are served by overhead travelling cranes. Seven of these cranes are of 30-ton and two of 50-ton capacity. Below nine of the bays, in which the underdrive presses are installed, is a continuous basement. In this basement is accommodated the elaborate conveyor system carrying the off-cuts from the presses above to the baling plant.

The steel store has a cantilevered, curved roof and is entirely free of support columns. This construction enables three 25-ton overhead cranes, spanning the full width of the store on runways 30 ft above floor level, to traverse the entire length of the building. Access for the heavy road vehicles bringing in steel is by four tunnel-type entrances. These are provided with inner and outer doors controlled by photo-electric equipment, so that only one door in a tunnel is open at a time and draughts and heat losses are avoided.

All steel is brought in by road transport. On an average, 720 tons of sheet and 170 tons of coil, are received each week. Approximately 40 per cent of this, representing off-cuts and trimmings, is returned in the form of baled scrap directly to the steel works by the delivery vehicles.

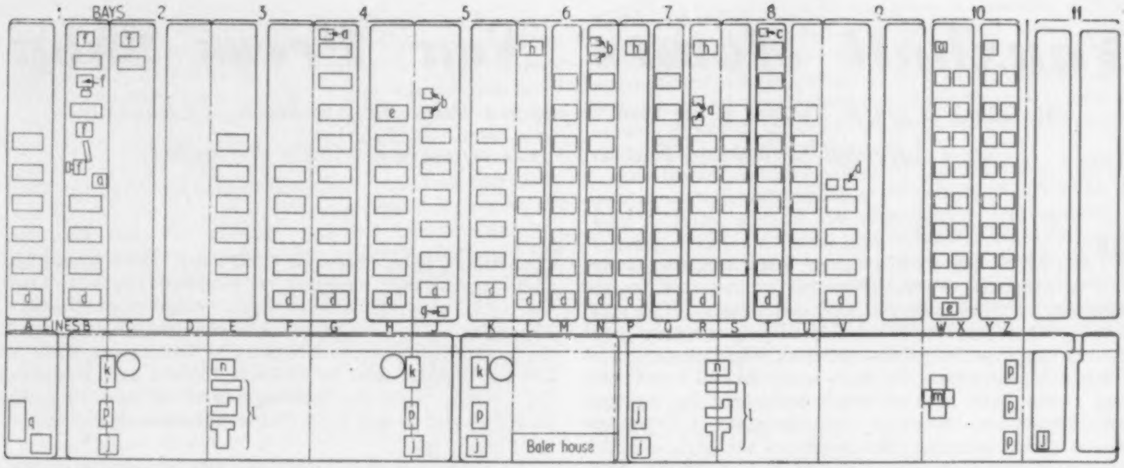
There are 25 lines of production presses, comprising in all a total of 222 machines. Of these, 118 are of the underdrive type, ranging in capacity up to 1,500 tons. These are British Clearing machines, supplied by Rockwell Machine Tool Co. Ltd., and the heaviest units weigh approximately 200 tons. The remainder are conventional overdrive machines; those of medium capacity by Wilkins and Mitchell and the smaller units by Hordern, Mason and Edwards. Six Clearing multi-spot welding presses are installed in the door lines and a seam welder for a preform operation in the front fender line.

Out of the total of 222 machines only three are of foreign manufacture. These are the Schuler, 9-station, transfer presses employed on the production of relatively small components requiring a series of press operations. A number of presses by various manufacturers, ranging up to 100-ton capacity machines by Craig and Donald, are transportable. They are lifted bodily and set down where required alongside a main press line for secondary operations.

A Bliss bedding and spotting press is installed in the bay reserved for die try-out and maintenance. In the steel store are Clearing overdrive blanking presses operated in conjunction with McKay coil feed lines, Wilkins and Mitchell blanking presses, Bronx flexing rolls, guillotines, and the duplex Lindemann hydraulic, scrap-baling plant.

Both shop and store are spacious and uncrowded by equipment. Throughout, the structure is painted in light, clear colours and all machines are painted a uniform grey. Natural lighting is excellent; the roof glass in the press shop represents 35 per cent of the floor area. Artificial lighting is by fluorescent tubes.

A typical heavy press bay. Line J producing front fenders and rear floor panels and line K for roof, floor, and hood panels



a *Hordern, Mason and Edwards press;* **b** *Craig and Donald press;* **c** *Hi-Ton press;* **d** *Clearing underdrive double-acting press;* **e** *Clearing top-drive, double-acting press;* **f** *weld press;* **g** *seam welder;* **h** *Schuler transfer press;* **j** *guillotine;* **k** *Bronx flexing rolls;* **l** *McKay coil feed;* **m** *slitting rolls;* **n** *Clearing top-drive press;* **p** *Wilkins and Mitchell press;* **q** *anti-drumming spray booth and oven*

Layout of press shop and steel stores; lines A to V—Clearing single-acting, underdrive presses; lines W to Z—Wilkins and Mitchell single-acting, overdrive presses; Bay 11—Hordern, Mason and Edwards inclinable presses

Cyclic production control

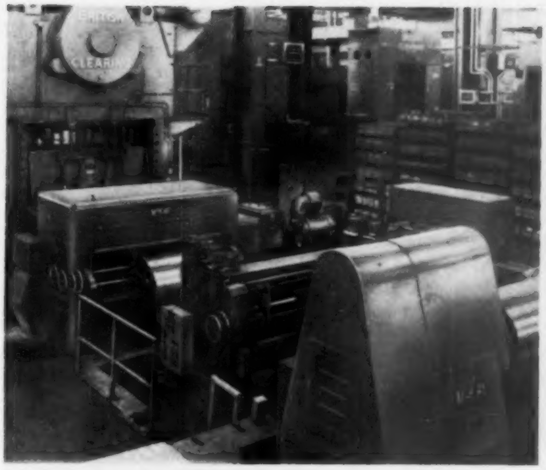
To attain flexibility in the large-scale manufacture of a complex, multi-component, end product in fluctuating demand, the most careful and far-sighted planning is required. Economy cannot be obtained by either speeding up or multiplying machines and operators but only by employing the most modern machines and handling and transfer equipment. In this respect, however, contradictory tendencies arise which militate against or may even cancel out specific advantages.

Already, machines and techniques have advanced beyond the individual requirements of what may be termed a "European" scale of production. Thus, the ideal of the continuous production and flow to the assembly line of a particular component becomes impracticable. Instead, an apparent reversion to batch production, transfer to store, and holding of stocks becomes necessary. A nice balance must be maintained if, while obtaining continuity and flexibility at the assembly line, down-time for tool changing, idle machines, transport and storage costs, and excessive stocks do not offset advantage gained in production.

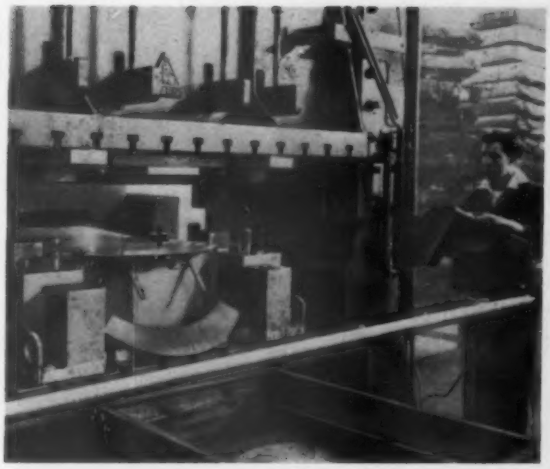
At Vauxhall production is arranged on a so-called "cyclic" system. The basic cycle period is determined by the assembly line requirements for twenty working days. This is estimated to cover all foreseeable contingencies of interruption of steel supplies, interruption of production, or sudden increase in demand, while spreading as widely as practical the period between changes of tooling on the machines.

The Sales Department issues a forecast of vehicle requirements for the ensuing six months, the monthly figures being cumulatively totalled. Each month a new forecast for six months is prepared, suitably amended by later knowledge of conditions, and taking into account any overrun or shortage arising from the previous month's operations.

View along gangway between press shop and steel store area. Line J, with preform seam welder in foreground, on the extreme left



One of the McKay coil-feed lines in the steel store



Blanking from coil on an automatically fed Clearing overdrive press

Production Control Department issues "daily build" figures to the assembly track and monthly production figures to the press shop.

The system is not applied inflexibly. Obviously, some reduction of the basic stock period can be made in the case of simple components that can be rapidly produced and some increase in the period is desirable for components of relatively complicated manufacture on a series of machines. Actually, specific instances may be cited where the period is as low as ten days and as high as twenty-five days. In any month the press shop is working on building up the stocks for the succeeding month's requirements. The stock position checked against the assembly line requirements at any time enables priorities to be fixed.

Presses are operated on the basis of 80 per cent efficiency. The time required for changing over the tooling of a complete line varies according to the size and weight of the dies and the number of presses involved. Typical times are 6 hours for a line of five of the largest presses (dies weigh from 35 to 40 tons and 50-ton cranes are used for lifting); 9 hours for a line of nine presses, and 2½ hours for a line of six of the smaller machines. No line is held in continuous operation on one component. Except in a case of emergency, one set-up only is made for the production of one month's requirement of any component.

Steel store

Delivery vehicles are unloaded by the overhead cranes and all steel is inspected and stamped before being accepted into store. The supplying steel mills maintain resident representatives to deal immediately with any question regarding the material that may be raised by the inspection department. Roll-over devices are installed so that parcels of sheet can be stacked in orderly fashion with the appropriate surface uppermost ready for delivery to the presses. Safety is assured by strict adherence to stacking regulations. Flat stock is stacked to a so-called "Plimsoll" line conspicuously marked on the walls about ten feet above floor level. Coils

General view of part of the coil and sheet steel store

are stacked vertically on edge in abutting lines, the end coils being positioned by angled wooden chocks. Coils in succeeding lines, to a limiting total height of four, are seated above pairs of coils in the line immediately below.

A variety of preliminary operations are performed in the store area. There are two lines for blanking or shearing from coil, each comprising a McKay feed line and a Clearing 300-ton overdrive press. The coil is deposited by the crane on to a carriage which lifts it to the self-centring, cone-type carrier from which it is drawn off by the feed rolls. Between the feed rolls stand and the measuring rolls stand, the material is looped in a pit 20 ft deep. At four different levels in this pit are sets of photo-electric cells controlling the operation of the feed-roll motor. When the loop of material interrupts the lowest beam, the motor is cut off. As the loop is taken up, the unmasking of the beams at succeeding levels runs the motor at relatively low, medium, and high speeds in order to maintain the loop before the measuring rolls. The feed system is linked under automatic control with the press and, in some operations, automatic stacking of blanks has been arranged. This equipment can handle coiled material up to 6 ft wide.

Four lines for dealing with sheet material each comprise a Schuler guillotine, a Wilkins & Mitchell 250-ton press, and Bronx flexing rolls. These flexing rolls are mainly used on the sheets for outer skin panels which subsequently have relatively complicated draws. Alongside the rolls are turntables to facilitate the operations of fork-lift trucks picking up the blanks or sheared stock.

Other equipment for preliminary operations includes a Weybridge gang slitting machine, a Schuler guillotine, three Wilkins & Mitchell 200-ton presses and thirteen H.M.E. presses. At the ends of the store building are the anti-drumming spray plant, opposite Bay 1 of the press shop, and a machine maintenance shop.

A salvage system is operated to utilize off-cuts of suitable size. Such items are given a stock number and serve as blanks or stock for smaller components.

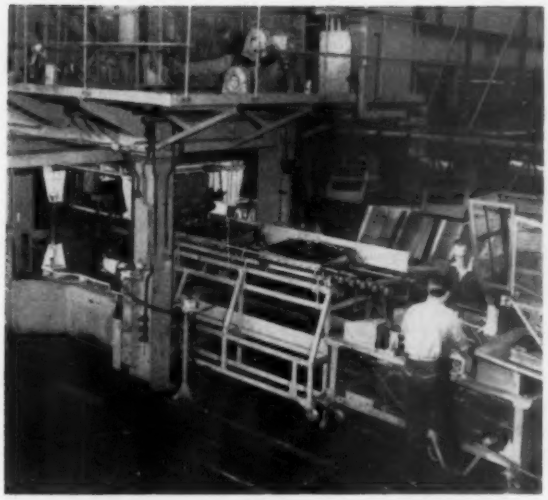
Press shop

With only two exceptions, all the presses are new equipment. Most of the machines, installed in eighteen lines in the first nine bays of the shop, are of the Clearing underdrive type. Bay 2 is unoccupied, providing a reserve area against future requirements, while Bay 9 is equipped for die proving and maintenance. The advantages offered by the underdrive design are fully exploited; indeed, the layout of the shop may be said to be based on this type of press. By comparison with the conventional overdrive design, underdrive presses have a clean configuration of low height above the production floor. Only the counterbalance cylinders are mounted in the crown of the press and there is no top hamper on crown or uprights to impede or obscure lifts by the overhead crane. Die setting is facilitated as a consequence of clearer access to the die area and the placing of the die is simplified since the bed level can be arranged flush with the floor. Much routine maintenance and minor overhaul work is removed from the production floor to the basement below. There is, thus, less likelihood of production being slowed down or even interrupted, or crane movement restricted, by the erection of ladders or stagings. Aisles are kept clear for the movement of stock, parts, or dies.

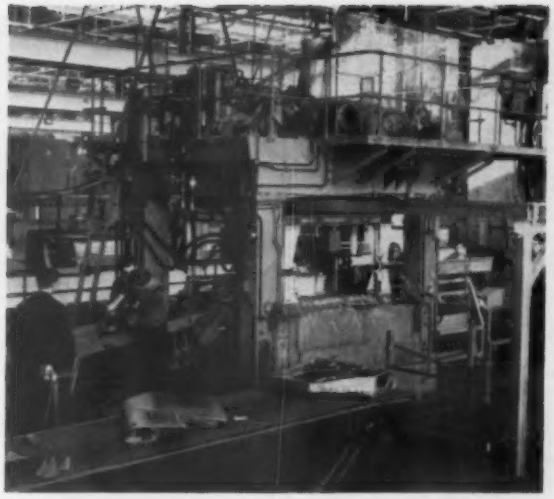
The presses are mounted on a framework of heavy I-section beams supported on massive concrete piers. On this framework the presses can be arranged with comparative freedom to suit a specific production programme. In the event of product changes necessitating a re-arrangement of the layout, machines can be re-sited without difficulty regarding foundations. Location of the drive gear below the press greatly facilitates maintenance or overhaul activities. Such work is not hindered by operations or traffic on the production floor and maintenance men or fitters work on the basement floor, or at a low elevation, with both hands free. Removal and re-assembly of equipment is effected by jacks and lift trucks, and by hoists attached to the girder framework. Dismantled parts may be laid out on the basement floor without obstructing aisles or impeding

Press line B for door panels. In the foreground, Clearing double-acting, 500-300-ton, 96 in × 60 in, blanking press

Clearing double-acting, 800-500-ton, 144 in × 84 in, blanking press at head of press line K for roof and floor panels



Clinching panels, shuttle-loading, and press-welding doors on line B



Delivery of doors from final press-weld machine in line B

activities on the production floor.

Bay 10 is occupied by Wilkins & Mitchell T.R. type, 200-ton, 36 in × 36 in, overdrive presses arranged in four lines. At the head of Lines W and X is a double-acting 48 in × 48 in Clearing overdrive press which serves both lines. A 75-ton H.M.E. inclinable press is installed at the end of Line W. All save one of the presses in Bay 11 are of the H.M.E. open-front, inclinable type in capacities of 40-, 55- and 75-ton. The exception, a Turner Bros. 30-ton machine equipped with a coil holder and a set of H.M.E. feed rolls, is used for progressive piercing of the Victor front grille. This component is subsequently flanged on one side on a folding machine.

It is not practicable to list all the components produced on all the lines, but the following examples, necessarily incomplete, are typical:

Lines A and B, eight different door assemblies—including multi-spot welding. Right-hand and left-hand, front and rear doors for the Victor and similar for the Velox and Cresta models.

Also two boot lids, for Victor and for Velox and Cresta.

Line G, wheelhouse panels, instrument panels.

Line H, sill panels.

Line J, front fenders—with preliminary seam-welding operation on blank.

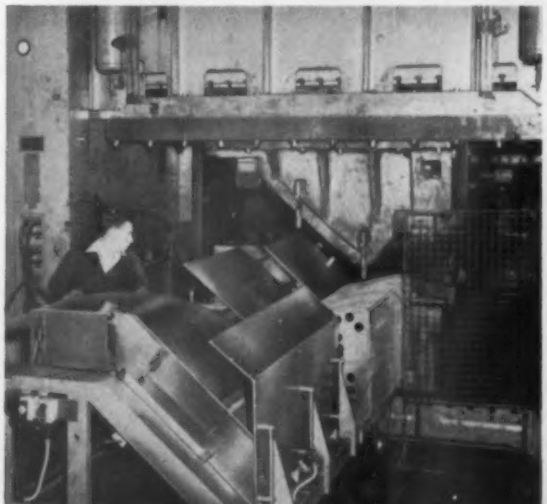
Line K, two roofs, two floors, two hoods.

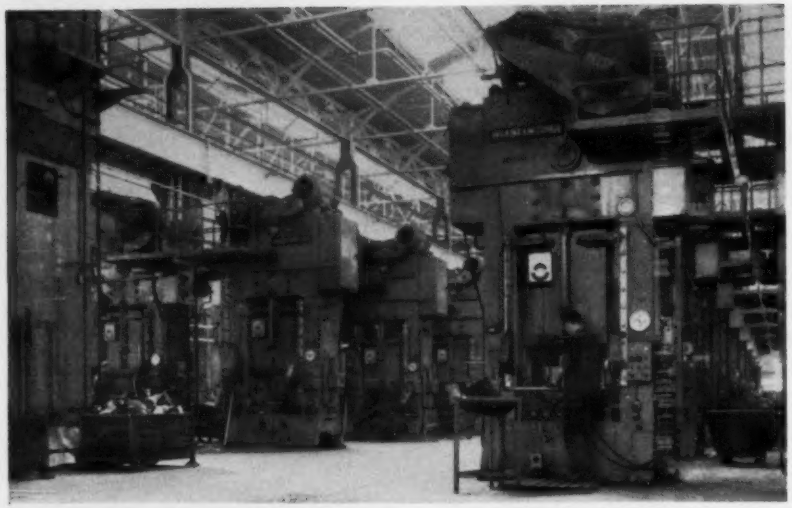
Press operations

Since Lines A and B are run in conjunction and include the assembly of the inner and outer panels produced, their operation is of special interest. For a right-hand front door a blank with a mitred corner for the inner panel is manually placed on a shuttle loader and fed to the 500-300-ton, 96 in × 60 in drawing press, A1. It is unloaded by an "iron hand" to a turn-over device which deposits it on rails and it is then manually loaded into a 400-ton 84 in × 60 in "spanking" press, A2, for a re-strike. An iron hand unloads it on to a portable band conveyor and it is then trimmed

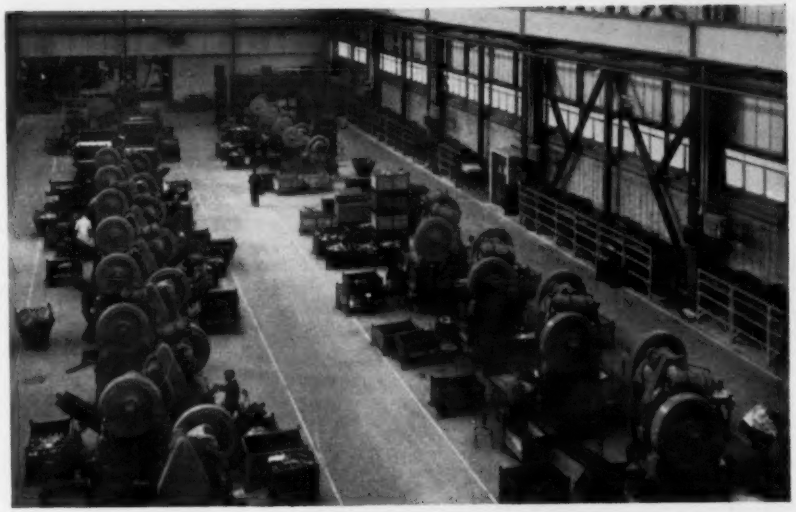
Shuttle loader for blanking press. Line B on door outer panels

Shuttle loader for blanking press. Line K on hood panels





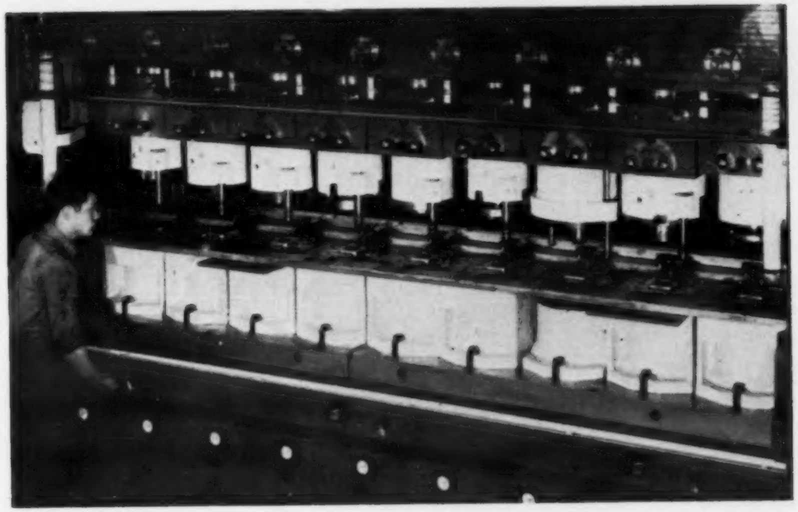
There are four lines of Wilkins and Mitchell 200-ton, 36 in $\times$ 36 in overdrive presses



Hordern, Mason and Edwards presses, 45-, 55-, and 75-ton, grouped in Bay 11

Schuler 300-ton transfer press, with coil feed rolls on left

View of tooling on Schuler press, showing grippers and transfer rails



Sequential operations in production of belt pulley on Schuler transfer press

and pierced in press A3, the tooling of which is furnished with scrap cutters. In press A4 one end wall is embossed, and special unloading equipment turns the panel horizontally through 90 deg. Then, in press A5 top and bottom walls are cam-pierced and in press A6 the two side walls are cam-pierced. Finally, it is off-loaded to a monorail conveyor for delivery to the assembly station. Presses A3 to A6 are of the same size and capacity as press A2. All are unloaded by iron hands and transfer between machines is by interpress band conveyors.

On Line B, equipped with similar-capacity machines, operations on the outer panel at presses B1 and B2 are draw and trim as on Line A. At press B3 three edges are flanged and at press B4 the upper edge is cam-flanged. Then on a H.M.E. 55-ton press the lock and handle holes are pierced. The panel is off-loaded to a monorail conveyor which carries it to the booth, located in the end of the steel store, where it is sprayed on the inner surface with an anti-drumming composition. After spraying, the panel makes three passes in a drying oven maintained at 270-300 deg F, and is then returned by the monorail conveyor to the assembly station.

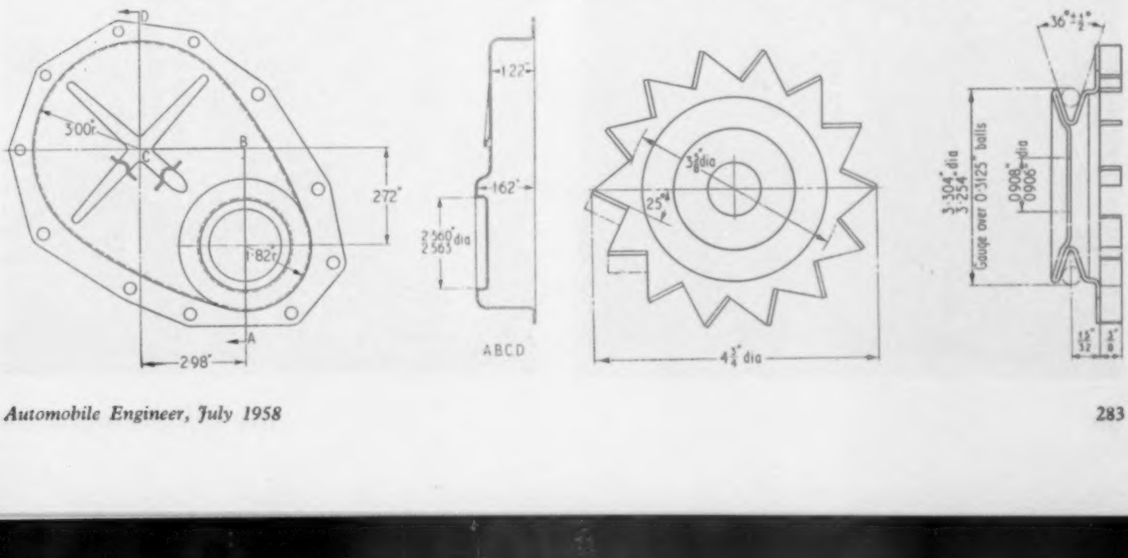
The assembly line is arranged as an extension of press line B. On a Federal twin-head, projection welder the hinge box is welded up and then attached to the panel corner extension pressing. This sub-assembly is then welded to the inner door panel on a Clearing EP.30 weld press. The panel is transferred on a belt conveyor to an EP.8 weld press and the arm-rest bracket and the trim-retainer strip are welded in and additional welds are made to complete the previous operation. Next it is conveyed to a 400-ton Clearing press for "spanking"—in particular to re-strike the flanges. It is unloaded by an iron hand and is then seated on the outer panel on a clinching fixture which is actuated pneumatically to secure the two panels together initially. A kick-out mechanism delivers the assembly to a shuttle loader which feeds it into a second EP.8 weld press. In this it is located and the panels finally secured together by fourteen spot welds. It is then passed on a roller track, fitted with sponge-rubber rollers to avoid marring the surface of the outer panel, to a 300-ton Clearing press for the last operation. This is the hemming of the two side, the bottom, and part of the top margins. Extraction of the door from this press is by an air-operated draw device with latching fingers. After inspection it is loaded on a cradle-type monorail conveyor for transfer to the finished assembly store located in a basement below the adjoining body-building shop.

The three Schuler transfer presses, supplied by Pearson, Panke Ltd., are sited at the ends of Lines L, P and R. They are of the latest type and are 9-station units of identical design. The tooling is interchangeable between machines and, consequently, the temporary withdrawal from service of one machine for maintenance work will not necessarily hold up production of any particular component. Each top toolholder on a press is furnished with individual, scale-indicated adjustment for height. Thus, once the specific tool height has been established, it is recorded to facilitate setting-up for subsequent production runs. Similar individual adjustment is provided for the ejector mechanisms. Adjustments for the main ram and also for the individual tool holders are motorized and controlled by push buttons.

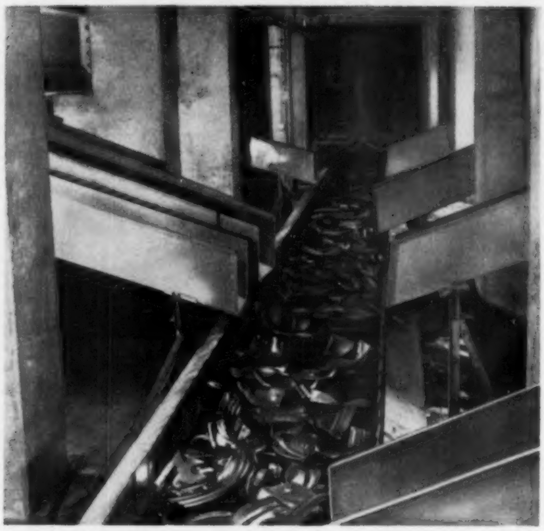
Since the presses must each be capable of producing any of the components required, it follows that for many of the simpler items produced not all the press stations are utilized. In such cases the working stations are spaced or grouped to avoid uneven loading of the ram and intervening stations are run idle. However, the operational flexibility ensured by a standardized installation remains the factor of major importance.

Providing batch quantities are large enough to justify

Two examples of components produced on the Schuler presses. Left: timing cover. Right: generator pulley and fan



Service station for battery trucks, equipped with Lansing Bagnall chargers. A complete battery change can be made in about two minutes



One of the scrap and off-cut conveyors in the basement. These feed the material to a duplex hydraulic scrap-baling press

runs of reasonable length and too much production time is not expended in frequent tool-changing, the use of transfer presses offers a high rate of production and substantial advantages as regards number of machines required, floor space occupied, and number of operators required. Since age-hardening of the material cannot occur between press stages the need for annealing operations is obviated.

With multi-stage operation the individual tools can be relatively simple and it becomes economical to produce as pressings from coiled strip stock, components that were formerly cast, forged, or fabricated. The generator pulley and fan, shown in line drawing, is a case in point. On this component lateral or radial run-out in relation to the bore is held to a limit of 0·015 in total indicator reading. The tooling for the production of belt pulleys includes rubber dies for the "bellying" operation which precedes the formation of the belt groove.

In the design of the press special measures are taken to ensure rigidity and the avoidance of deflection in the structure under load in order to maintain the essential precision of movement and accuracy of register. The narrow table, uprights and the crown form a single box frame. The deep ram is guided in four long gibs in each upright and is suspended at each side by a pitman from an eccentric shaft mounted directly over the upright. In operation the applied pressure is taken as tension by the uprights and the crown is relieved of all major stresses. The last of the working stations is located in the right-hand column to ensure positively spring-free conditions for final sizing operations.

Transfer of the work from station to station is by means of a parallel pair of rails fitted with appropriate gripper elements. These rails move inwardly to grip the workpieces and then stroke to the right to progress the parts. After location, the rails open to release the parts, and then return ready to repeat the operation. As a safety measure each gripper device incorporates a feeler switch and in the event of a

Part of 30,000 ft^3 door and boot lid assembly store in basement

Part of panel store at press shop level. Note special pallet racks

part being incorrectly located the press is immediately stopped. Indicator lights show which device is at fault.

Each of the three presses is equipped with an automatically controlled coil feeder of Schuler manufacture. It embodies a duplex coil stand of the expanding mandrel type, mounted on a carriage arranged for lateral displacement so that one coil can be feeding while another is being loaded. From the coil the material is drawn through a set of variable-speed levelling rolls which deliver it in a loop to measuring rolls built on the left-hand upright of the press. The speed of the levelling rolls is controlled by means of a feeler arm engaged by the depending loop of material and the drive is electrically interlocked with the press to cut out immediately the press is stopped. As it enters the measuring rolls the stock is wiped clean and as it is delivered to the blanking station it is automatically lubricated.

Hub covers furnish a typical example of the components produced on the Schuler presses. Only seven of the nine stations are used and the sequence is as follows:

1. Blanking
2. Drawing
3. Trim on outside diameter
4. Flange wiped down and re-drawing to concentric profile
5. Idle station
6. Idle station
7. Cam pierce two holes in side rim
8. Pierce centre hole and form curl on lower edge
9. Finalizing curl

Stock and parts handling

Overhead monorail conveyors are employed only for the doors and boot lids produced on Lines A & B. On other lines movement of blanks and pressings is by battery-operated fork-lift trucks of various types and sizes. Maximum flexibility is maintained by palletizing all components for transfer into and out of the storage areas. For the smaller range of pressings, Lansing Bagnall pedestrian-controlled trucks are employed. These are compact, easily manoeuvrable vehicles having a capacity of 2 tons over a lift of four inches, and incorporate a regenerative braking circuit which extends the working period between necessary recharges of the battery. A rider-controlled version is used on the longer hauls for which its higher operating speed makes it more suitable. Actually, throughout the Luton plant, more than 150 of these vehicles are engaged on various duties.

Although Lansing Bagnall trucks are equipped with built-in chargers, by means of which the batteries can be recharged at the correct rate by plugging into any convenient mains supply point, the shift system worked at Luton does not leave sufficient free time for this method of charging. To avoid having a large number of vehicles out of service at a time, spare sets of batteries are maintained at a central charging and service station. A row of battery chargers, specially designed to match the 48-volt truck batteries, is used, and the provision of a loading platform at the correct height allows a complete battery change to be made in about two minutes. An automatic cut-out prevents the truck from being operated when the voltage falls below a safe level.

Scrap collection and baling

The outpouring of off-cuts, punchings and trimmings from a plant of this size raises problems of collection and handling of considerable magnitude. At Luton the waste material from all the heavy, underdrive presses is handled smoothly and expeditiously by a network of apron-slat conveyors in the basement. As already mentioned the tools of the trimming presses are furnished with scrap cutters, and the cut scrap cascades down inclines and chutes on to conveyors arranged parallel to the press lines. The conveyors run in the same direction as the press lines and at their end points deliver into similar conveyors running left and right at right angles and delivering into the main conveyor below Line M presses. This traverses the width of the press shop and the steel store and delivers into the hopper of the Lindemann duplex baling press.

In the loading dock the scrap bales are handled by a lifting magnet on a travelling hoist

Bales of scrap weigh from 4 to 5 cwt and are compressed to a standard width, though length and height can vary. They are delivered through a tunnel by means of a 126 ft conveyor of the twin-chain and pusher type, to the loading dock in a covered building separate from the main block. An electro-magnetic hoist lifts the bales on to the road transport vehicles.

J. B. DUNCAN

As the last pages of this number close for press, we have learnt with deep sorrow of the death of James Brownlie Duncan, editor of this journal since 1st November, 1952. Mr. Duncan, who had had several serious illnesses over the past few years, was taken gravely ill at his home on Friday night, June 27th, and died the following afternoon in St. Thomas's Hospital. He would have been 61 years of age on the 24th of this month.

Automobile engineers the world over will join with us in sympathy to his widow.

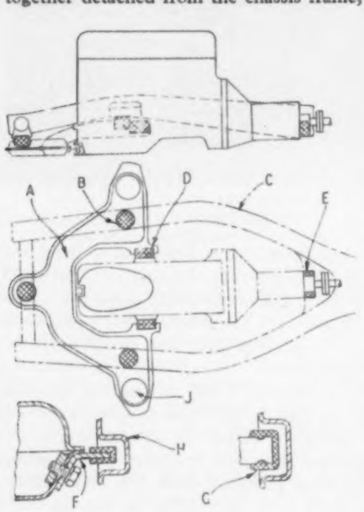
Mr. Duncan was appointed to the editorial staff of the *Automobile Engineer* on 26th June, 1939, primarily to cover new plant, tools and production methods. After service in the first World War—he enlisted in the Seaforth Highlanders at the age of 17 and won the Military Medal—he was trained in shipbuilding. Later he became a research engineer with the Coventry Chain Co. Ltd., with responsibility for development work on product design and production methods. Next he was in charge of the inspection, from new material and tool-making to finished products, at the Renold and Coventry Chain Company. After a spell as works manager at the Suffolk Iron Foundry, Ltd., he set up as a consultant and finally, as stated, joined the *Automobile Engineer*, a sphere in which his meticulous attention to detail, the immense fund of knowledge he acquired and his delightful personality were to make him the friend and counsellor of automobile engineers almost without number. For ourselves, we have lost a very dear colleague—an editor to whom we and the journal owe much.

CURRENT PATENTS

A REVIEW OF RECENT AUTOMOBILE SPECIFICATIONS

Suspension of engine and gear unit

Where an engine unit is resiliently suspended from an auxiliary frame resiliently suspended from the chassis frame, the possibility exists that the engine may oscillate too violently when the vehicle encounters an uneven road surface at high speed. Furthermore, when the engine unit and the auxiliary frame are together detached from the chassis frame,



No. 777226

there is a danger that the unit may tilt excessively in relation to the auxiliary frame. The invention overcomes these disadvantages.

Auxiliary frame A is detachably and resiliently suspended at three points B from the forked chassis frame C. The engine and gear unit also has a three-point resilient suspension; at two forward points D from the auxiliary frame and at the rear from a central point E from the chassis frame.

To prevent excessive movement of the engine unit in a vertical plane, a short arm or tongue F is secured to the auxiliary frame. The free end of this arm enclosed in a rubber cap G enters, with both vertical and lateral clearance, a recess H formed in the front end of the crankcase sump. Clearances are arranged to permit only an average range of movement on each side of the normal static position.

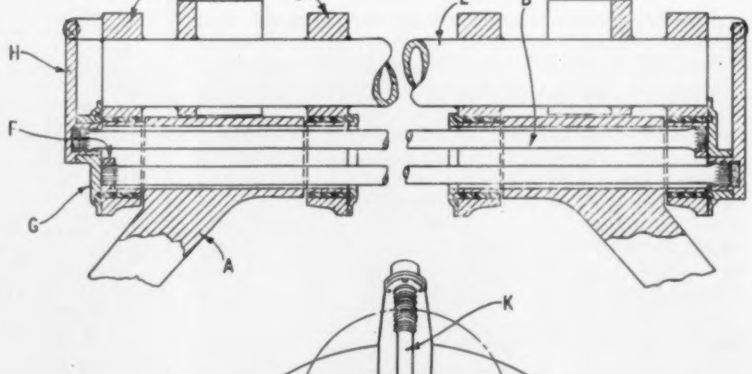
Wheel suspension units are mounted on the limbs of the auxiliary frame, with the helical springs seated at J. *Patent No.* 777226. *Daimler-Benz A.G.* (*Germany*).

Torsion bar front suspension

In this independent front wheel suspension, the wheels are mounted on trailing arms A pivoting about a common transverse axis on the vehicle frame. The hubs of the arms are hollow to accommodate a pair of torsion bars B, each extending completely across the frame. At each side of the frame are brackets C and D, welded to a tubular cross member E, formed at their rear ends with transversely aligned bores. These bores are fitted with rubber bushings to receive the hollow hubs of the arms A.

The outboard end of the hub is partially closed by an integral flange F formed with a splined hole to receive the end of its respective torsion bar. An approximately semi-circular aperture remains, to furnish a clearance passage for the other torsion bar. A cover plate G, bolted to the outer face of bracket C, has a radially offset boss bored to receive the splined hub of lever H, in which the end of the second torsion bar is anchored. The angular position of the lever H can be varied to adjust the riding height of the vehicle by means of an abutment bolt J mounted in a lug on bracket C.

It will be seen that the bars B are subjected to combined torsional and bending stresses. The suspension is completed by tubular shock-absorber struts K linking the ends of arms A to a frame structural member. *Patent No.* 775740. *Ford Motor Co. Ltd.*

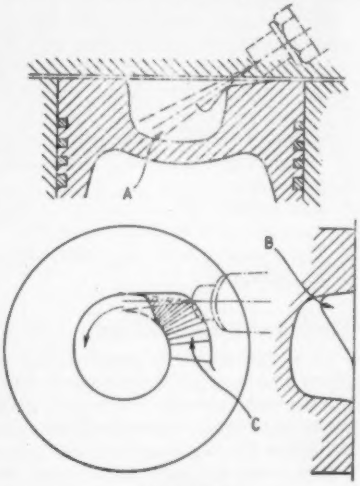


No. 775740

Piston-crown swirl chamber for diesel engine

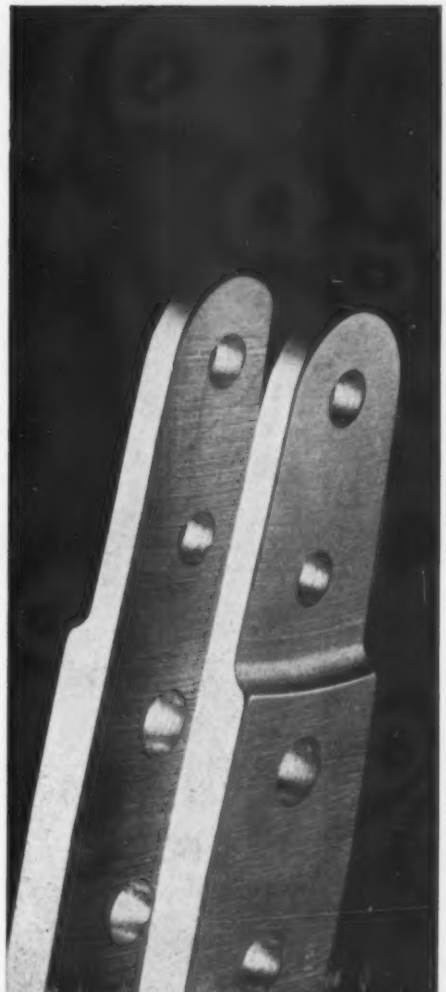
It is the aim of this design to impart to the combustion air an intensive turbulent flow about the piston axis with the minimum loss of energy. Viewed in plan, the wall of the centrally disposed combustion chamber A constitutes a single complete turn of a spiral. The helical surfaced portion C of the piston face, running along that part of the wall having the smaller radius, terminates in trough-shaped recess B merging with the wall at its greatest radius. During the compression stroke, air displaced from the cylinder flows into the chamber by way of surface C and recess B, creating an intensive swirl in the chamber and obviating the violent transverse eddies over the lip of the chamber usually engendered by "squish". Mounted at an angle in the cylinder head, the atomizer injects a single fuel spray along recess B, tangential to the chamber wall.

The helical surface C guides the air positively to the fuel spray. When starting up, the air velocity is relatively low, due to the slow rotational speed of the engine. The fuel spray is not deflected on to the cold wall of the chamber and, thus, starting is facilitated. As the engine speed increases,

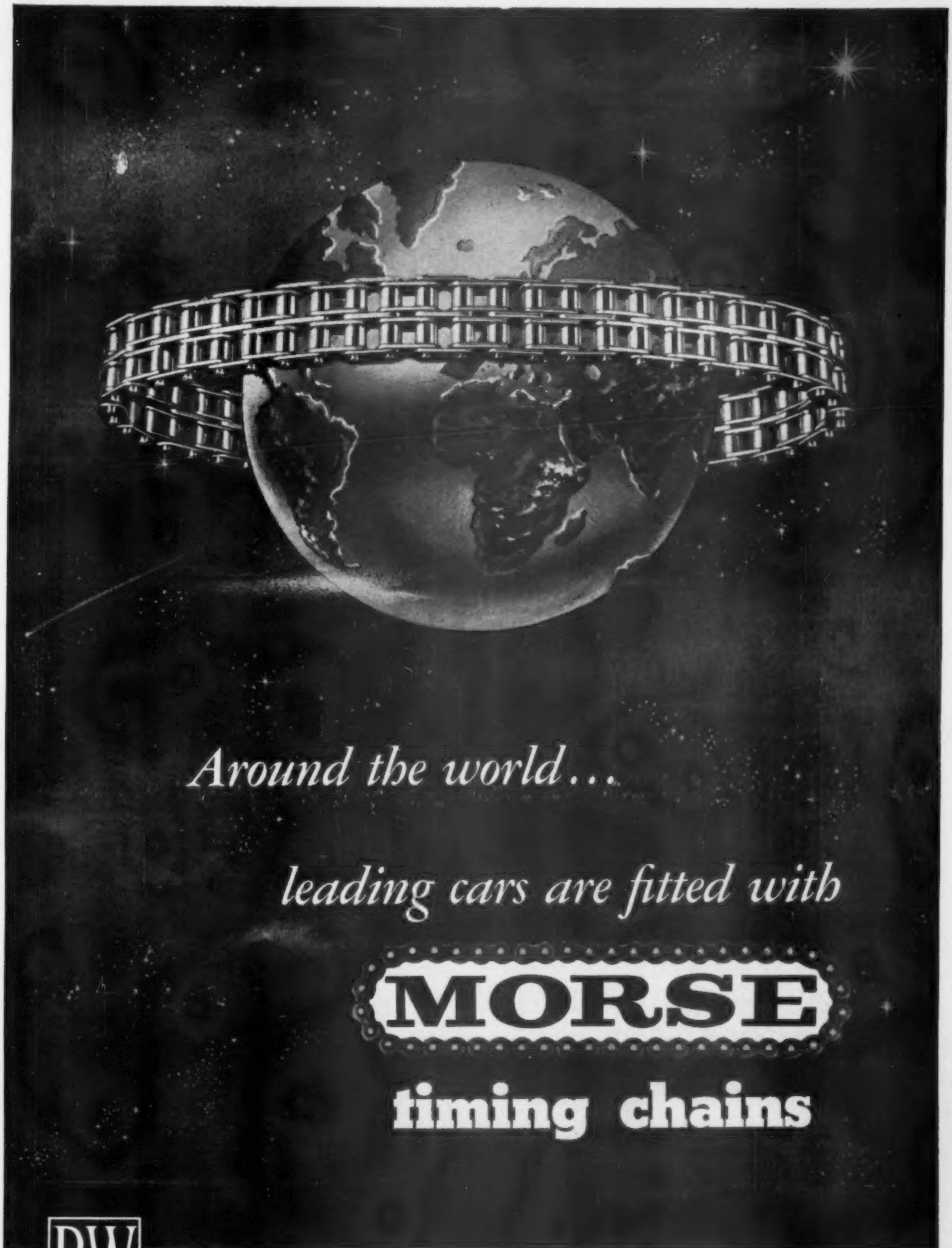


No. 777531

however, the air velocity is raised and the full spray is deflected on to the then hot wall of the chamber. Here the still liquid fuel is vaporized and the combustion conditions are improved. *Patent No.* 777531. *Steyr-Daimler-Puch A.G.* (*Austria*).



Alloy steel fork, plated with Kanigen all over for corrosion resistance, for use in aircraft applications.

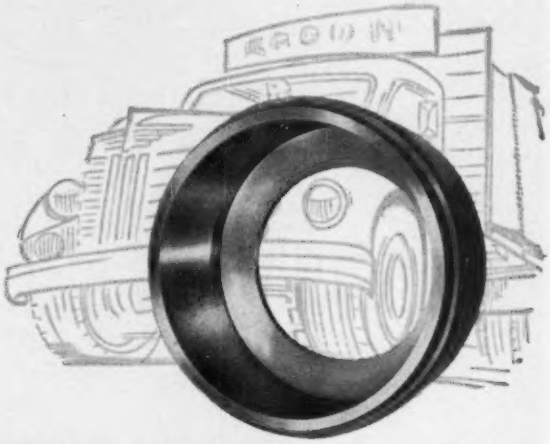

Around the world...
leading cars are fitted with
MORSE
timing chains

FOR SOUND CASTINGS

CONSULT

We are manufacturers of high duty iron castings for the motor car and allied trades.

Manufacturers of CROMLEX High Duty Cast Iron—Ideal for Brake Drums.



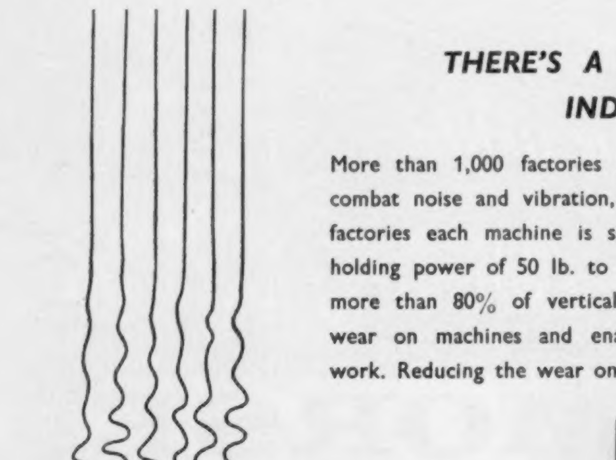
TRIPLEX FOUNDRY LTD

UPPER CHURCH LANE, TIPTON, STAFFS. *Telephone Tipton 1293 (4 lines)*

HIGH DUTY IRONFOUNDERS

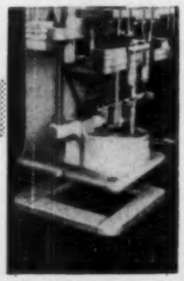
A Member of the Triplex Foundries Group

THE SIGN OF A GOOD CASTING



THERE'S A SOUND BARRIER IN INDUSTRY, TOO!

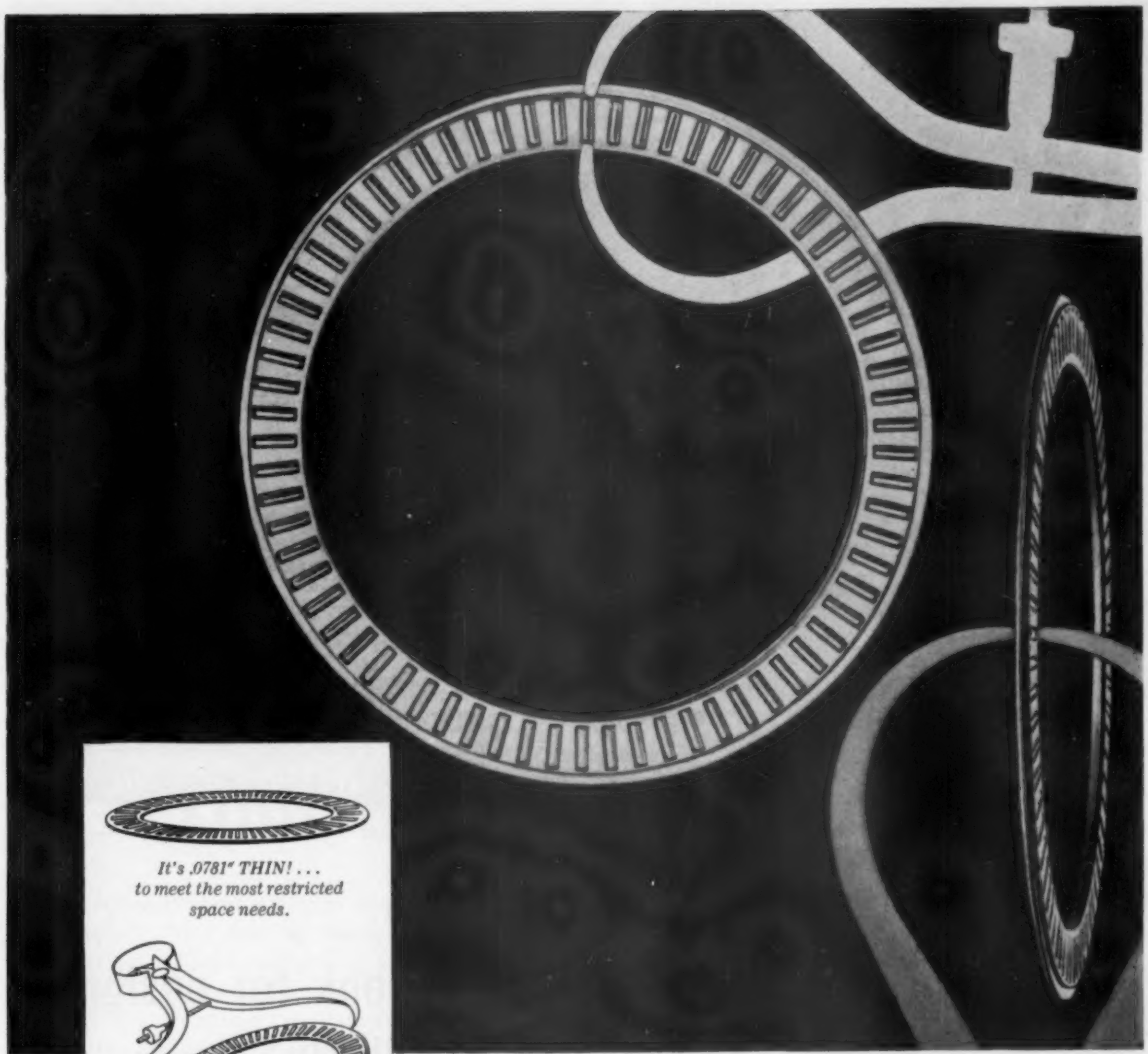
More than 1,000 factories now use the Croid-Cooper method to combat noise and vibration, the arch-enemies of efficiency. In these factories each machine is simply stuck down on a felt base with a holding power of 50 lb. to the square inch, a felt base that absorbs more than 80% of vertical vibration in some cases, reducing the wear on machines and enabling them to produce more accurate work. Reducing the wear on operators' nerves, too. With the Croid-Cooper method bolting and grouting are things of the past and layouts can be changed over-night.



Send for further details today.

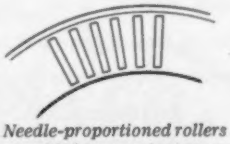
CROID 65
MACHINE FIXING GLUE
COOPERS
FELT

COOPER & CO. (BIRMINGHAM) LTD. BRYNMAWR
BRECONSHIRE
Telephone: BRYNMAWR 312.



It's .0781" THIN! . . .
to meet the most restricted
space needs.

OD is much smaller,
for given shaft size, than
other types of thrust bearings.



Needle-proportioned rollers
provide large contact area
in minimum cross section.

By every measure . . . ideal for compact thrust applications

Whether you gauge its value in compactness, high anti-friction efficiency, high thrust capacity, or low unit cost, you will find the new Torrington Needle Thrust Bearing measures up ideally to your needs.

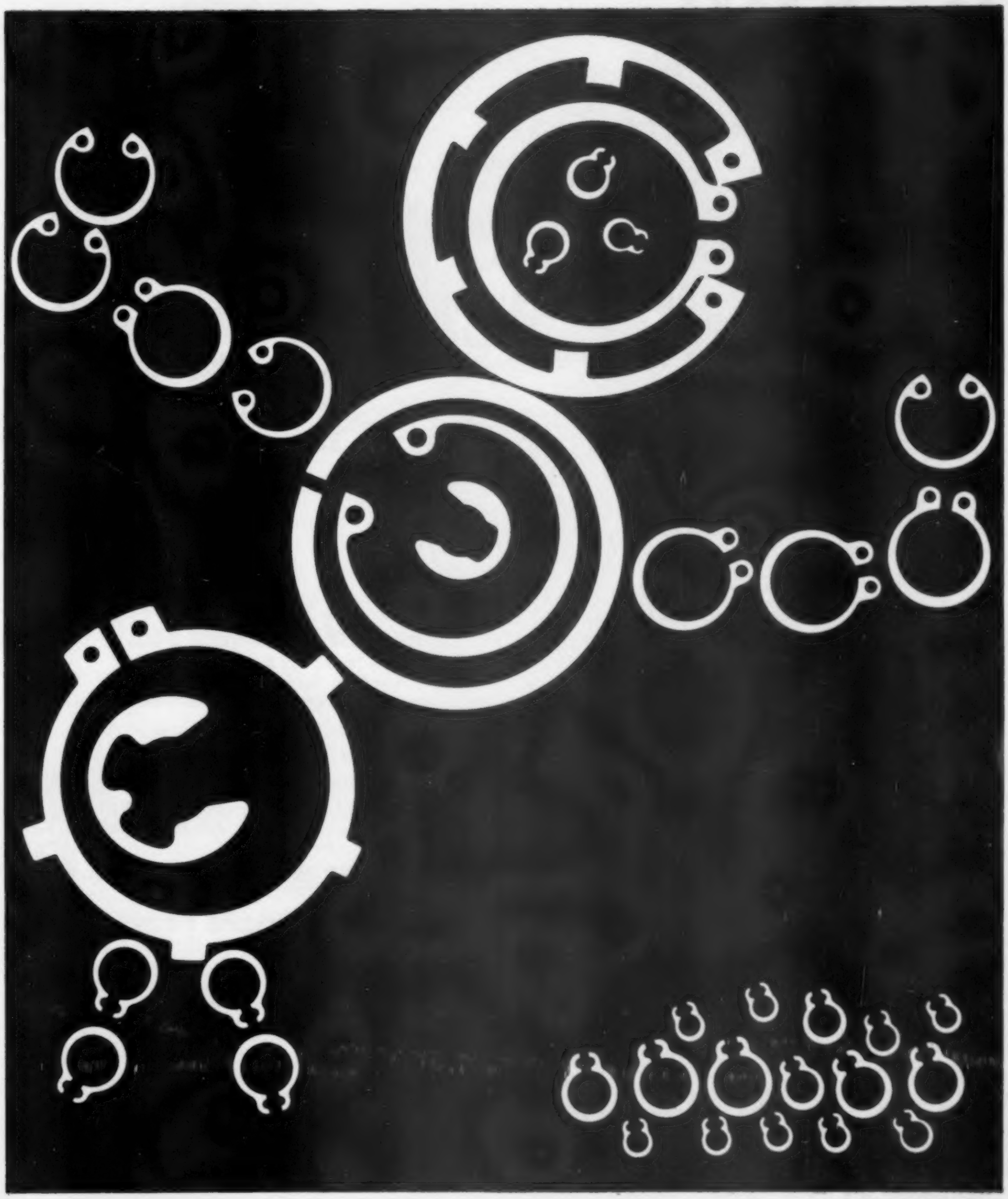
This needle-type bearing is designed specifically for thrust loads in restricted space. It may run directly on adjacent hardened and ground surfaces or on thrust races which may be simply and economically produced to suit individual design requirements. Used alone, or in combination with Torrington radial type Needle Bearings, the Needle Thrust Bearing finds wide use in many applications including steering gears, hydraulic pumps, tractor bolsters, bevel and worm gear boxes, governors, outboard motors, 2-cycle engines, washing machines, power tools, torque converters, and automatic transmissions.

Please write to our Bearings Division for catalogue 758 and call upon our services for assistance in design.

ΦB/58/E

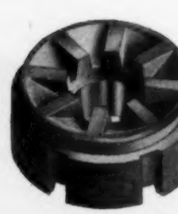
TORRINGTON BEARINGS

THE TORRINGTON COMPANY LTD. BEARINGS DIVISION: TORRINGTON AVENUE, COVENTRY
LONDON & EXPORT OFFICE: 7-10 ELDON STREET, E.C.2. GLASGOW OFFICE: 14 MOIR STREET C1



GEARS

and even more complex shapes produced without machining with DURASINT



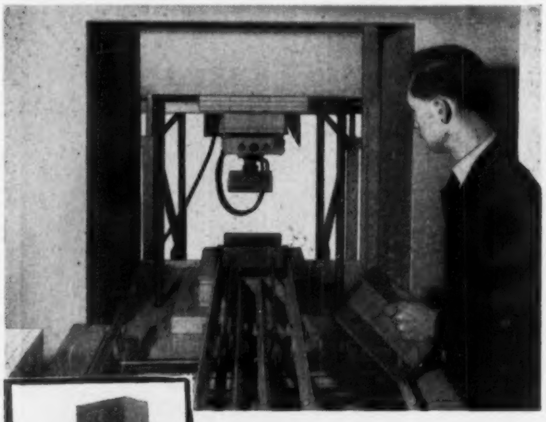
Think of the savings you would make if you could eliminate machining! Impossible, you might think? No, not a bit of it. For so great are the advances made in recent years in powder metallurgy that it is now possible to produce a vast range of pieces **in their finished state!**

Already many manufacturers dealing in quantity production are taking advantage of these new techniques. And at Sintered Products Limited, still further applications are being found daily to help industry achieve a faster, more economic production of parts. Perhaps they could help you?

Our Representative will gladly call to discuss your particular problems. Or write for the DURASINT booklet giving you full details to:

Sintered Simon says
Save on SINTERED

Sintered Products Ltd.
(One of the Sheepbridge Engineering Group)
Hamilton Road, Sutton-in-Ashfield, Notts.
Telephone: Sutton-in-Ashfield 590   Telegrams: Sintered, Sutton-in-Ashfield



UDAL Press Unloader and Loader give four-fold saving.

1. LABOUR SAVING The Unloader either replaces an Operator at back of Press or enables the front Operator to concentrate solely on feeding.

2. TIME SAVING The Loader permits continuous feeding with the minimum of Operator movement. The two units together on a Press running at 15 r.p.m. will give a production rate of 800 per hour with one Operator. With manual loading, two Operators would do well to reach 360 per hour.

3. VERSATILITY Easy change over from one job to another. Mounted on castor wheels with built-in height adjustment, the Units can be rolled clear for toolsetting or moved to another machine and re-set within minutes.

4. ECONOMICAL COST Relative to the savings in production costs from the use of these Units, their prices including installation, are attractive.

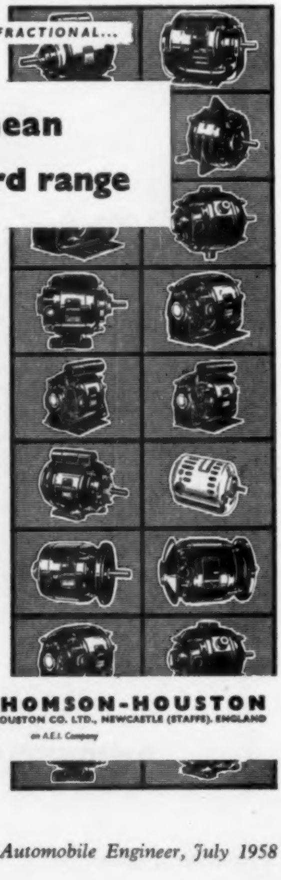
J. P. UDAL LTD., Interlock Works, Court Road, Birmingham 12.

UDAL

Telephone: CALthorpe 3114.

WHEN THE DRIVING POWER IS **FRACTIONAL...**

What we mean by a standard range

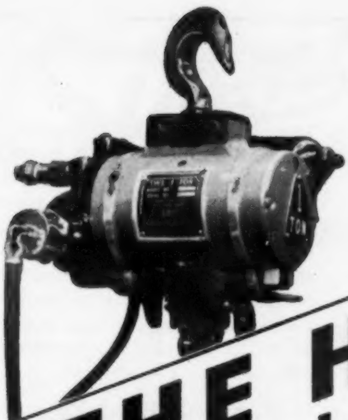


BTH fractional horse-power motors are made in a wide range—a wide range of mountings, a range of different enclosures and fittings, and a range of special-purpose types of which those illustrated form only a small selection. There is virtually a standard motor for any job. They are made in sizes from one-thousandth horse-power. There are base mountings or cradle mountings; spigot, resilient, ring or automatic tension mountings; horizontal, vertical and 'any-angle-you-want' mountings. There are AC motors, DC motors; geared units, auto-belt tension units; flame-proof or drip-proof enclosures; for high, low or medium speed applications at all standard voltages. Designers of machinery therefore have not only the widest field of choice but can take full advantage of standardisation and interchangeability.

A 'Fractional' Advisory Service For Designers
The long experience of BTH engineers with fractionals for all purposes is at the disposal of designers. Consultation at the drawing board stage can save you time and trouble. When you are planning new machines or modernising existing installations, get in touch with BTH.

B T-H

BRITISH THOMSON-HOUSTON
THE BRITISH THOMSON-HOUSTON CO. LTD., NEWCASTLE (STAFFS), ENGLAND
an A.E.I. Company



THE HOIST for the loading bay

"ARO-BROOMWADE"

For the loading bay there is no better Hoist than the "ARO-BROOMWADE" Pneumatic Hoist—measuring only 10½" x 5" diameter—highly efficient, economical and easily handled, it can lift weights up to 1 ton.

Available also in ½ ton capacity.

A wide range of stationary air compressors is available for the supply of compressed air.

Write for full details.

"BROOMWADE"

AIR COMPRESSORS & PNEUMATIC TOOLS

Your Best Investment

BROOM & WADE LTD · P.O. BOX No. 7 · HIGH WYCOMBE · ENGLAND

Telephone: High Wycombe 1630 (10 lines) Telegrams: "Broom", High Wycombe, Telex

577 SAS.

P. M. HELDT *S.A.E.*

explains principles and practice in the American automobile industry

The Automotive Chassis

Without Power Plant. 3rd Edition. An authoritative exposition of American automobile practice. This valuable book deals comprehensively with all aspects of frames, springs, axles, wheels, tyres, drives, steering gear, brakes, universal joints and differential gears. 8½"×5". 616 pp. 400 illustrations. 75s. net. By post 76s. 9d.

Torque Converters or Transmissions

5th Edition.

Describes the principles and operation of all modern American types of automobile clutches and gearboxes. Friction, automatic, hydraulic and electric clutches are dealt with, and the chapters on power transmissions include electric, hydrostatic, hydro-kinetic, automatic and pneumatic systems. 8½"×5". 476 pp. 285 illustrations. 65s. net. By post 66s. 9d.

High-Speed Combustion Engines

16th Edition.

First published in 1912, this well-known standard textbook covers American practice in the most comprehensive manner. All aspects of the subject are covered in great detail, including the design, production and testing of all types of petrol engines. Profusely illustrated, this book is a complete treatise on current American practice. 8½" × 5". 776 pp. 521 illustrations. 100s. net. By post 101s. 9d.

High-Speed Diesel Engines

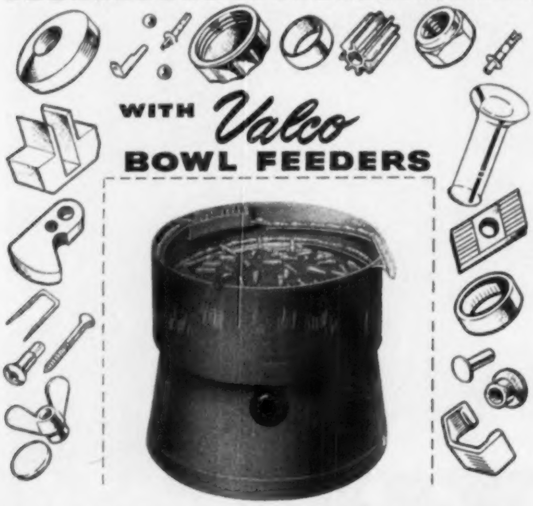
7th Edition.

For Automotive, Marine, Railroad and Industrial Use. A guide to American practice in diesel engines of the automotive type. Design, fuels and fuel injection equipment, supercharging, lubrication, operation and maintenance are among the subjects dealt with, and there is also a chapter on gas turbines. 8½" × 5". 472 pp. 288 illustrations. 60s. net. By post 61s. 9d.

From leading booksellers, or direct from:—

ILIFFE AND SONS LIMITED - DORSET HOUSE - STAMFORD ST., LONDON, S.E.1

AIDS TO AUTOMATION



For the feeding of small components to automatic machines of all descriptions. Components are fed out in single file in correct alignment to be received by the machine. Rate of feed variable, easily set to suit the particular job.

Feed tracks can be designed to handle awkwardly shaped components in the manner required.

Bowl Feeders available in a range of sizes from 6" to 36" diameter.

Send for leaflet No. 160.

MANUFACTURERS OF VIBRATORY FEEDING AND ELEVATING EQUIPMENT FOR POWDERS, GRANULES, ETC.

VALLEY PRODUCTS (LYE) LTD

VALLEY ROAD . LYE . STOURBRIDGE

Collier's

WORM DRIVE HOSE CLIP

Phone: DIDcot 2046.

COLLIER & COLLIER LTD.

NORTH MORETON, DIDCOT, BERKS. - - ENGLAND

ARE YOU re-designing?

Very often our design engineers can improve the structural form of your castings, and at the same time reduce complexity and cut costs. That way you'll get cheaper, more sound castings of unequalled quality and accuracy. This technical teamwork can tackle *all* your casting problems —and *solve* them.

BIRSO TRADE MARK

CASTINGS FROM A FEW OUNCES TO 10 TONS

in phosphor-bronze, gun-metal, aluminium-bronze, manganese-bronze, and light alloys. Precision machined bushes and bearings. Specialists in high-tensile aluminium-bronze castings, centrifugal-cast wheel blanks, and chill-cast rods and tubes.

Continuous cast phosphor-bronze bars up to 12 feet lengths.

NON-FERROUS CASTINGS

HIGH-DUTY IRON CASTINGS—PRECISION MACHINED BUSHES & BEARINGS

T. M. BIRKETT, BILLINGTON & NEWTON LTD

HANLEY AND LONGPORT, STOKE-ON-TRENT, ENGLAND

***Head Office:* HANLEY, Phone: Stoke-on-Trent 22184/5/6/7. LONGPORT, Phone: Stoke-on-Trent 87303**

Britain's Roman Highways

ERMINE STREET

When the Romans invaded Britain they were faced with the problem of creating suitable highways which would give their legions access to the ends of the country. Of the many they built the most important and famous were the Four Ways; Watling Street, The Fosse, Ermine Street, and Icknield Street.

Ermine Street, the third of the Four Ways, was a direct road from London to Lincoln and on to Wintringham on the Humber. By means of a ferry it made communication with Brough and York. North of Lincoln another road diverged from it Westwards to Doncaster. Near London the Enfield Highway is part of the old road.

In Lincolnshire, Ermine Street is known as a 'ramper' or 'The Old Street.' Like Watling Street and the Fosse Way, Ermine Street appears to have been paved from end to end.

Today the historic highway has become part of the main road transport system of the country. This system is about to be modernized so that the transport of the country may continue to flow evenly.

* Richard Berry have been actively concerned in improving the passage of vehicles over the roads of Britain and the World for over a hundred years. Today the tradition of their skilled craftsmen is being kept alive by the products of their new road spring factory.

R B TRADE MARK

write for your copy of

"LEAF SPRING DESIGN"

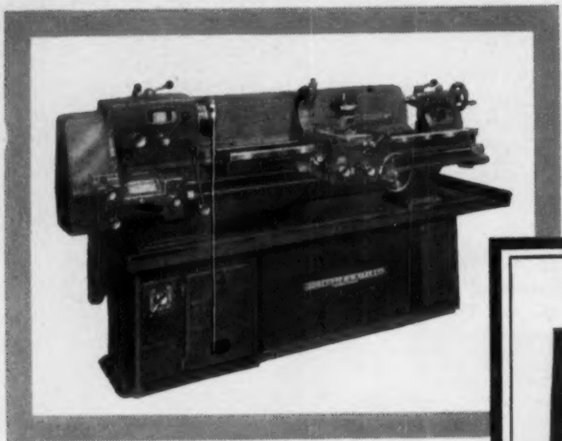
by Alan Hodgson

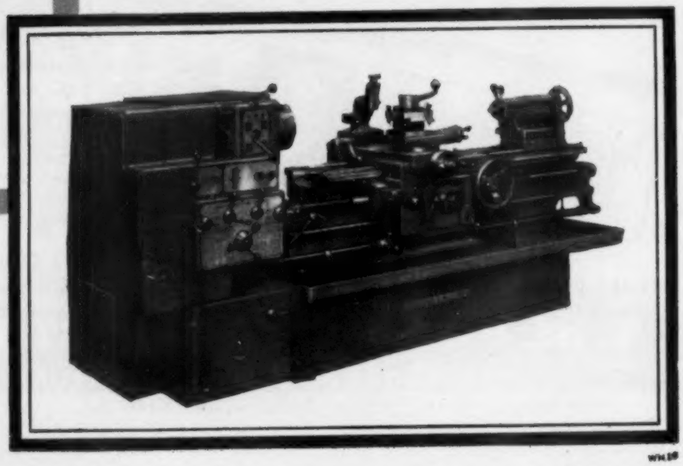
CARRY ON BERRY

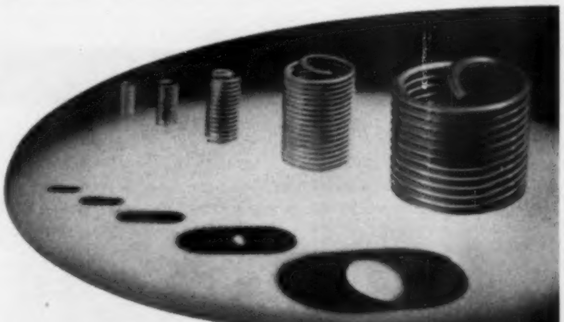
RICHARD BERRY & SON, WEST BROMWICH, STAFFS.

Tel: West Bromwich 1766-7-8.

ONE OF THE BROCKHOUSE COMPANIES

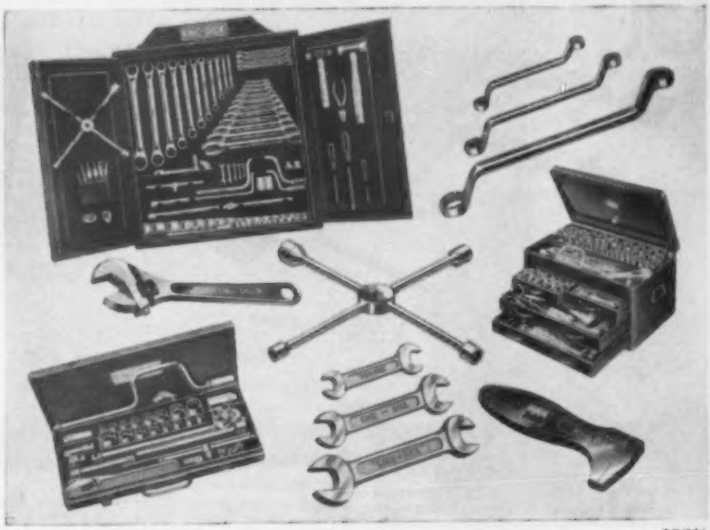






TOP DOG

WITH PRECISION MADE **HAND TOOLS**



. . .Equip with KING DICK, the hand tools with the tradition of constant quality, superb design and excellent finish.

From the largest Constructional Engineering Tools to the smallest B.A. Socket Set, KING DICK inspection routine ensures the correct combination of selected steels, perfect balance and precise accuracy.

BUY BRITISH !

ABINGDON KING DICK

ABINGDON WORKS · KINGS ROAD
TYSELEY · BIRMINGHAM · 11

RE/201

ANNOUNCING

A MAJOR BRITISH DEVELOPMENT IN BEARING TECHNOLOGY

Bearings are now available which operate

- **Completely dry or in most liquids**
- **Under heavy loads or high speeds**
- **At temperatures from —200°C. to +250°C.**
- **In the presence of dust or abrasive particles.**

Nearly 1,000,000 of these bearings are already running in a wide range of mechanisms from gas turbines to textile machinery.

Three materials with different functions have been developed to assist those who wish to design machinery that avoids the use of lubricants.

DU Thin steel strip with a porous bronze coating impregnated with a mixture of a fluoro-carbon (P.T.F.E.) and lead. Supplied in flat lengths or as finished bushes and thrust washers in a range of sizes. Available at low cost from stock.

DQ A fluoro-carbon (P.T.F.E.) strengthened with special fillers and supplied in bars and tubes. A range of diameters is available from stock. Non-standard or irregularly shaped dry bearings can be simply machined from this material.

DM A process of applying an adherent layer about .002" thick utilising the bearing properties of a combination of fluoro-carbon and molybdenum disulphide to the bearing surface of any ferrous part. Such parts must be sent to us for process.

The basic material we use in all these bearings is the "Fluon" brand of Polytetrafluoroethylene supplied to us by I.C.I. Ltd.

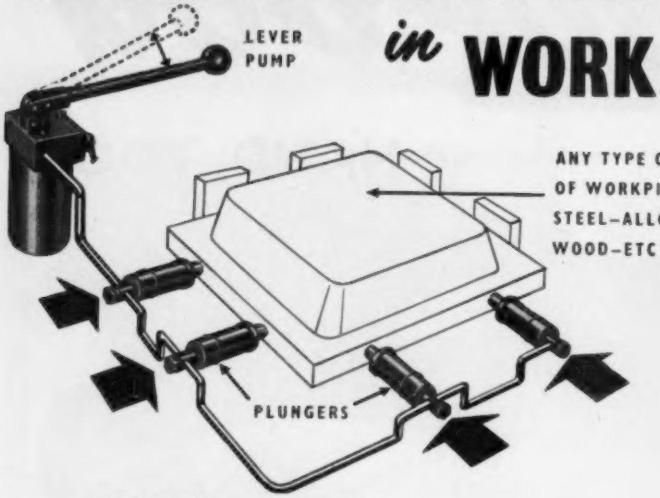
DU DU is a new development based on our original DP material which it now supersedes, and provides three times the load/speed carrying capacity of its predecessor.

GLACIER

Information sheets giving guidance on use, maximum loads, speeds etc., are available free on request. TEL: PERIVALE 6611. TELEX: GLAMET, WEMBLEY.

THE GLACIER METAL COMPANY LIMITED · ALPERTON · WEMBLEY · MIDDLESEX

BIGGEST TIME & MONEY SAVER *in* WORK CLAMPING



NEWTON
TRADE PRODUCT MARK
Hydraulic
CLAMP
SYSTEM

These plungers can exert a pressure of from 800lbs—4 tons.

Can be used in groups of any number.

Maximum pressure with smallest plungers

Easily adaptable to any fixture design

Simultaneous plunger action

Minimum floor to floor time.

Send today for illustrated brochure H/243

POWER JACKS LIMITED
MAYLANDS AVENUE HEMEL HEMPSTEAD

---

an egg or an omnibus!

PARKER PACKING

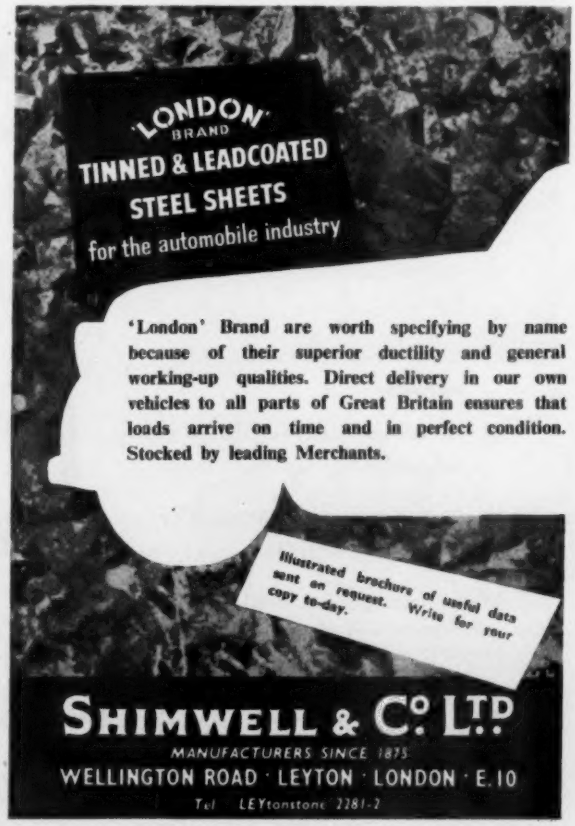
will see it across the World!

Situated with direct access by water to all the Thameside docks . . . equipped for heavy handling—with acres of storage space . . . Parker Packing also offers the "know-how" and conscientious care which provide maximum protection for valuable exports.

Parker Packing is a highly specialised organisation, developed and trained for the specific purpose of packing for export with effective economy.

Get a fresh slant on your packing problems by calling in

PARKER PACKING
—PROTECTS YOUR EXPORTS!
The Parker Packing Co. Ltd.,
Victoria Wharf, Grove Street, Deptford, S.E.8.
Telephone TIDeway 6511



Classified Advertisements

RATE 4d. PER WORD · MINIMUM 4/-

Each paragraph charged separately. Box number 5 words plus 1/-

Advertisements for the August 1958 issue should be to hand not later than first post 28th July

(No responsibility accepted for errors)

FOR SALE

100 BAYS of brand new adjustable Steel Shelving, 72 in. high × 34 in. wide × 12 in. deep, stove enamelled dark green, sent unassembled, six-shelf bay, £3 15s. 0d. Sample delivered free. Quantity discounts. N. C. Brown Ltd., Eagle Steelworks, Heywood, Lancs. Tel. 69018. [5613

PATENTS

THE Proprietor of British Patent No. 723,773, entitled "Fuel Injectors for Internal Combustion Engines," offers same for licence or otherwise to ensure practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, 14 E. Jackson Blvd., Chicago, Illinois, U.S.A. [5643

THE Proprietor of British Patent No. 722,859, entitled "Automobile Parking Elevator," offers same for licence or otherwise to ensure practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, 14 E. Jackson Boulevard, Chicago 4, Illinois, U.S.A. [5645

THE Proprietor of British Patent No. 727863, relating to fuel atomizers for I.C. engines, desires to enter into negotiations with firms in the United Kingdom with a view to the commercial exploitation of the invention. Enquiries should be addressed, in the first instance, to Philip S. Allam, 2 and 3 Norfolk Street, Strand, London, W.C.2. [5648

THE Proprietor of British Patent No. 727082, entitled "Improvements in a Combined Control Device for the Brakes and Accelerator of a Motor Vehicle," offers same for licence or otherwise to ensure practical working in Great Britain. Enquiries to Singer, Stern & Carlberg, 14 E. Jackson Boulevard, Chicago 4, Illinois, U.S.A. [5649

THE Proprietors of British Patent No. 706,933, for "Machine Tools for Machining Brake Drums and the Like," desire to enter into negotiations with a firm or firms for the sale of the patent, or for the grant of licences thereunder. Further particulars may be obtained from Marks & Clerk, 57 and 58 Lincoln's Inn Fields, London, W.C.2. [5652

TUITION

A.M.I.M.I., City and Guilds, A.M.I.Mech.E., etc., on "No Pass-No Fee" terms. Over 95% successes. For details of Exams. and Courses in all branches of Auto., Aero., Mechanical Eng., etc., write for 144 page handbook — Free. B.I.E.T. (Dept. 643), 29 Wright's Lane, London, W.8. [5553

JOHNSON & JOHNSON
(Gt. Britain) Ltd.
PERMACEL TAPES DIVISION

Applications invited from experienced Salesmen with good contacts within the Motor Trade covering spray painting shops and Motor Garages engaged in Motor Body painting work.

The successful applicant will be given a thorough training and then will be appointed to an area in London and South of England.

His efforts will be supported by means of advertising and sales promotional work. He will be paid a good salary immediately on starting with commission and expenses. Car will be provided. Applicants, who should be between 25 and 35, should write in confidence to the General Sales Manager, Permacel Tapes Division, Johnson & Johnson (Gt. Britain) Ltd., Slough, Bucks.

Samuel Marsden & SON LTD
Altrincham Street, Manchester 1

London Rep: W. Kelway-Bamber & Co. Ltd., Room 7, 70, Victoria Street, London, S.W.1 Telephone: Abbey 6860

N. E. Coast: Fasteners Ltd., 2 Hall St., Barnard Castle, Co. Durham. Tele.: Barnard Castle 3172.

dmSM.38

North Staffordshire Technical College.
Stoke-on-Trent.

Principal: H. W. Webb, O.B.E., D.Sc., F.R.I.C., M.I.Chem.E., F.I.Ceram.

ENGINEERING DEPARTMENT

1. Applications are invited for the appointment of full-time ASSISTANT LECTURER, GRADE "B" to commence in September or as soon as possible afterwards. Applicants should be able to teach the subjects of the City and Guilds course in AUTOMOBILE ENGINEERING PRACTICE. They should hold some recognized degree, certificate, or other professional qualification. Salary in accordance with the Burnham Scale for Grade "B" Assistants: £650 × £25—£1,025, with full additions for training, graduate qualifications and experience, which may in some cases bring the maximum up to £1,225 per annum.*

2. Applications are also invited for the post of full-time ASSISTANT LECTURER GRADE "A" to teach MOTOR VEHICLE WORK to commence in September. Applicants should be capable of teaching to at least the standard required by the City and Guilds for Motor Vehicle Mechanic's Work and should preferably possess a full technological certificate in Automobile Engineering Practice. Salary in accordance with the Burnham Scale for Grade "A" Assistants: £475 × £25—£900 per annum, with full additions for training, graduate qualifications and experience, which may in some cases bring the maximum up to £1,100 per annum.

Applications for both the above posts giving details of age, qualifications, training and experience, together with the names of two referees, should be sent to the Principal immediately. Applicants should state clearly for which post they are applying.

H. DIBDEN,
Clerk to the Governors.

For automatic and capstan precision parts—in any metal—to your own specifications . . . *consult the specialist machinists.*

I.F.V., D.A.I., D.I.Arm., A.R.B. Fully Approved.

REPETITION PARTS *from the* BAR *by* . . . .

MCL

Telephone: Broadwell 1115 (4 lines) and 1757

M.C.L. & REPETITION LTD.
POOL LANE · LANGLEY · BIRMINGHAM

SERVICES OFFERED

DESIGN Office Capacity available for Mechanical and Production Engineering. Special Machines and Projects. Norris Brothers Ltd., 53 Victoria Street, London, S.W.1. Tel.: Abbey 5444. [5610

SITUATIONS VACANT

AUTOMOBILE Engineer with flair for writing, for the editorial staff of "Motor Transport" and "Bus & Coach." The ideal man: young, enterprising and ambitious; familiar with current goods and passenger vehicle design, including diesel engines; and will preferably have worked in a road transport fleet engineering department. Good salary and prospects. Write in confidence to Philip Edwards, Editor, "Motor Transport," Dorset House, Stamford Street, London, S.E.1. [5650

SITUATION WANTED

A.M.I.MECH.E., now design section leader, with present company 10 years, seeks position with wider opportunities. Experience in complete chassis design, experimental and project work. Good practical background and administrative ability. Age 44. Present salary £1,350. Southern England preferred. Box 6954, c/o Automobile Engineer. [5651

BOOKS

TWENTY-TWO bound volumes of "Automobile Engineer," 1939 to 1957 inclusive, also 1933 and 1934, also composite volume interesting selections over 1927-1938. £100 or best offer. Carriage paid U.K. Woolfenden, Crossmead, Bromham, Bedford. [5647

SALES SUPERINTENDENT
DAVID BROWN INDUSTRIES LIMITED

wish to appoint a Sales Superintendent who will be responsible for all sales in the U.K. of the

AUTOMOBILE GEAR DIVISIONS

of the Company which are concerned with the production of gears and transmissions for the Commercial Vehicle and Automobile Industry. The essential qualifications are a sound engineering background, proved sales ability in a similar position and established high level contacts in the industries concerned. The position is pensionable and permanent and will be located at the Company's main gear works in Huddersfield. The Salary envisaged will be commensurate with the duties and responsibilities of the position.

Applications giving full details of education and positions held should be addressed to the Chief Personnel Superintendent, David Brown Industries Ltd., Park Works, Huddersfield.

COTTON BAGS
FOR SPARE PARTS Etc.
Walter H. Feltham & Son Ltd
Imperial Works, Tower Bridge Road
Telephone: HOP 1784 LONDON, S.E.1

---

new and revised edition

AUTOMOBILE BRAKES AND BRAKE TESTING

By Maurice Platt. 2nd edition. Nowadays, every motorist or individual concerned with motor-vehicles should have some knowledge of brake mechanism. This new edition incorporates the technical information required by the automobile engineer with the fundamental principles that the general reader needs to know. Specially compiled so that even readers without a technical training can understand it, this new edition is an essential book for the driver and mechanic. Pitman's "Automobile Maintenance" Series. 15/- net.

PITMAN

Parker St., Kingsway, London, WC2

INDEX TO ADVERTISEMENTS

Printed in Great Britain for the Publishers, ILIFFE & SONS LTD., Dorset House. Stamford Street, London, S.E.1, by James Cond Ltd., Charlotte Street, Birmingham 3. "Automobile Engineer" can be obtained abroad from the following: AUSTRALIA & NEW ZEALAND: Gordon & Gotch Ltd. INDIA: A. H. Wheeler & Co. CANADA: The Wm. Dawson Subscription Service Ltd.; Gordon & Gotch Ltd. SOUTH AFRICA: Central News Agency Ltd.; Wm. Dawson & Sons (S.A.) Ltd. UNITED STATES: Eastern News Co. Entered as Second Class Matter at the New York, U.S.A., Post Office.

There's power in the atom . . .

Sand cast in aluminium alloy. Diameter 49¾", maximum depth 19¾", fettled weight 536 lb.

and strength in BIRMAL

This is a very special casting. Somewhere behind the creation of atomic power it has an important part to play in the production and processing of fuel. Birmal excel in large complex castings of this kind for both the atomic and the engineering industries. A background of more than half a century's experience lies behind their choice of materials and processes. The benefits of this knowledge and experience are available to all industries where the quality of castings count.

You get more than a casting from BIRMAL

Birmingham Aluminium Casting (1903) Co. Ltd.

BIRMID WORKS SMETHWICK BIRMINGHAM 40